Engineering Library
APR 13 1920

VOL. VI NO. 2

THE JOURNAL OF THE SOCIETY OF AUTOMOTIVE ENGINEERS

FEBRUARY 1920

SOCIETY OF AUTOMOTIVE ENGINEERS INC.
29 WEST 39TH STREET NEW YORK

Non-Gran
Bronze Bushings

Long Life in a Bearing Bronze

is not secured by this or any other chemical specification

NOR will physical tests, in combination with chemical analyses, predetermine the service life of a bearing bronze.

The sure way to get a dependable, long-life, wear-resisting bronze is to do as Baldwin Locomotive, General Electric, Curtiss Aeroplane, Locomobile and hundreds of other big industrial and automotive manufacturers do and that is—specify Non-Gran on your prints. Then you know that you will get a bearing bronze of proved dependability made by bearing bronze specialists who have made nothing else for over ten years—a bearing bronze of unvarying uniformity, free from flaws, sand holes, air or gas pockets—a bronze that will give your machines longer service life by postponing bearing renewals and overhauling dates.

Bear this in mind: You cannot "write" long life into a bearing by means of a specification. But, by writing "NON-GRAN" on your blue prints you get the long life Non-Gran insures—not once but every time.

We'll gladly send you copies of the Non-Gran booklets and a list of prominent manufacturers using Non-Gran. Address:

AMERICAN BRONZE CORPORATION
BERWYN **PENNSYLVANIA**

DISTRICT OFFICES

760 Tremont Bldg., Boston, Mass.
481 People's Gas Bldg., Chicago, Ill.
Pacific Coast—Allied Industries, Inc. - - - 979 Minna St., San Francisco, Cal.
340 Mateo St., Los Angeles, Cal.
3010 Woolworth Bldg., New York, N. Y.
340 Leader-News Bldg., Cleveland, O.
1252 1st Avenue, So., Seattle, Wash.

Foreign Distributors

Foreign Dept., Automotive Products Corporation, 3010 Woolworth Bldg., New York, N. Y.
Stanley J. Watson, Sheen Road, Richmond, London, England.
Aktiebolaget Galco, 18 Norra Bantorget, Stockholm, Sweden.
Cycle Motor Trade Supply Co., Capetown and Durban.
Henderson & Co., Johannesburg, South Africa.
John O'Neill Pty., Ltd., Sidney, Australia.
Suers. de L. Villamil and Co., Porto Rico.
Simson and Nielson, Copenhagen, Denmark.

HIGH SPEED
NON-GRAN
BEARING BRONZE

THE JOURNAL OF THE SOCIETY OF AUTOMOTIVE ENGINEERS

Vol. VI February, 1920 No. 2

1920 Standards Committee

A SUBSTANTIALLY complete list of the personnel of the 1920 Standards Committee is given in this issue of THE JOURNAL so far as divisions are concerned that the Council of the Society has decided to continue or establish. It will be recalled that the 1919 Standards Committee consisted of eighteen Divisions. There will be this year twenty-one or more Divisions at work actively on the 186 subjects before the Standards Committee, most of which have been formally assigned for disposition.

The Farm Engine and Lighting Plant Division has been divided into the Isolated Electric Lighting Plant Division and the Stationary Engine Division. A Silent Chain Division, in addition to the Roller Chain Division, has been established.

Under the guidance of Chairman B. B. Bachman, the Standards Committee will be governed by practically the same regulations as were put into effect last year. These regulations have been prepared for issuance to members in data sheet form for filing in the *S. A. E. Handbook.*

PROPOSED WORK OF THE DIVISIONS

The Aeronautic Division, which has had before it for some time a long program of work, largely resulting from war conditions, will hold early sessions with a view to formulating further standards and recommended practices suitable for use in connection with the production of airplanes and lighter-than-air craft. The proceedings of the Division will be participated in by representatives of Government departments, bureaus and committees.

The Marine Division has an ambitious program, many of the engineers and executives connected with the production of motor boats feeling that every effort must be made to give their field the benefit of such further standardization as will facilitate more extensive use of motor boats under conditions of greater satisfaction and less cost.

The Stationary Engine and the Isolated Electric Lighting Plant Divisions are working in close cooperation with the National Gas Engine Association.

The Tractor Division will proceed as heretofore. A Sub-Division of this Division has been appointed to work jointly with committees of the National Implement & Vehicle Association and of the American Society of Agricultural Engineers in the formulation of Agricultural Equipment Standards under a plan recently proposed and put into effect. The working principle of this plan is that Agricultural Equipment Standards may be established by the affirmative action of any two of the three bodies named, that is, the S. A. E., the National Implement & Vehicle Association and the American Society of Agricultural Engineers. It is appreciated that there is much overlapping work in this connection, as involved by the tractor on the one hand and the implements with which it is used on the other. While a very large amount of dependable data were developed in farming during the days of the horse, and much of this is applicable to power farming, it is clear that the need is great for further research in connection with the problems of power haulage of farm implements.

The work of the Standards Committee Divisions in general applies to all the various automotive apparatus for the most part. There is no passenger-car division of the committee as such, although there are a Truck Division and a Motorcycle Division, in addition to divisions bearing the names of specific automotive apparatus as mentioned above.

VALUE OF STANDARDIZATION

With regard to the essentiality of intelligent standardization, few of intimate knowledge of the automotive industry need information or assurance. In a summary of advantages to be gained, made recently by D. K. Boyd in relation to building construction, it is made clear that standardization in its application to the erection and equipment of modern structures is as essential to successful results as is the standardization of time, weight and measures. Attainable results are the elimination of a tremendous amount of individual effort, time and expense in specification preparation and detail drafting, a saving of lost motion and waste in the production and application of materials and the manufacture of equipment. Production can be kept well in advance of demand. Needless variation in types, sizes, patterns and finishes, together with unnecessary carrying of large stocks, is greatly reduced. Appropriate use is made of materials in conformity with their known characteristics, including limitations; and of proved methods of manufacture. Knowledge of suitableness makes possible reduction of cost of construction and of maintenance. The wider use of best adapted materials improves working and living conditions, insures a high uniform degree of safety and efficiency during and after construction, makes possible closer cooperation between engineers and their associates engaged in the production, furnishing and installation of the materials. The needs or desires of all, not the whims or caprices of some, are the great desiderata. Needless variation in type or pattern causes useless ex-

pense and many difficulties in merchandising, fills shelves and stores with surplus stock, and contributes to the high cost of building.

A standard is "that to which one turns," and by which judgment is given. A basic function of the Society of Automotive Engineers is to reduce cost to the producer and to the consumer, making feasible at the same time better products.

Personnel

B. B. Bachman, *Chairman*

AERONAUTIC DIVISION

Henry M. Crane, *Chairman*

Commander A. K. Atkins
Col. C. de F. Chandler
Major V. E. Clark
C. H. Day
D. W. Douglas
W. L. Gilmore
Major G. E. A. Hallett
J. L. Harkness
Commander J. C. Hunsaker
C. B. Kirkham
G. C. Loening
F. R. Porter
W. T. Thomas
R. H. Upson
C. M. Vought
C. F. Willard
P. W. Wittemann

AUTOMOBILE LIGHTING DIVISION

Henry M. Crane, *Chairman*

Paul F. Bauder
William T. Jones
W. A. McKay
C. A. Michel
Edward S. Preston
C. H. Sharp

BALL AND ROLLER BEARINGS DIVISION

W. C. Baker
W. L. Batt
George R. Bott
E. R. Carter, Jr.
D. F. Chambers
F. C. Goldsmith
F. W. Gurney
E. Nides
H. J. Porter
R. G. Schaffner
W. R. Strickland
R. E. Wells

Sub-Division on Roller Bearings

W. C. Baker
T. V. Buckwalter
L. R. Heim
A. M. Laycock
R. G. Schaffner
C. W. Spicer

R. E. Wells

ELECTRIC TRANSPORTATION DIVISION

Commercial Vehicle Sub-Division

Edward L. Clark, *Chairman*

P. M. Holdsworth
Frank E. Queeney

Charles A. Ward

Passenger Vehicle Sub-Division

George M. Bacon
Karl Probst

Battery Sub-Division

Bruce Ford, *Chairman*

Walter H. Bancroft
Walter E. Holland

Industrial Truck and Tractor Sub-Division

J. M. Breitenbach
Willard C. Brinton
Frank L. Eidman
Thomas T. Fauntleroy
Albert H. Gfrorer
M. J. Harkless
P. W. Saitta
M. S. Towson

H. W. Woodward

ELECTRICAL EQUIPMENT DIVISION

A. H. Timmerman, *Chairman*

F. W. Andrew
Joseph Bijur
R. J. Broege
Roger Chauveau
W. A. Chryst
Bruce Ford
W. A. Frederick
C. F. Gilchrist
W. S. Haggott
Victor W. Kliesrath
T. L. Lee
B. M. Leece

A. D. T. Libby

ENGINE DIVISION

H. C. Snow, *Chairman*

R. J. Broege
J. B. Fisher
W. A. Frederick
E. G. Gunn
A. F. Milbrath
D. G. Roos

E. O. Spillman

IRON AND STEEL DIVISION

F. P. Gilligan, *Chairman*

J. S. Ayling
C. N. Dawe
E. L. French
Walter F. Graham
H. L. Greene
C. G. Heilman
E. T. Ickes
L. O. Koven
M. B. Morgan
John H. Nelson
G. L. Norris
J. H. Parker
W. C. Peterson
C. F. W. Rys
H. J. Stagg
R. A. Townsend

H. F. Wood

ISOLATED ELECTRIC LIGHTING PLANT DIVISION

L. S. Keilholtz, *Chairman*

G. M. Gardner
E. T. Gilliard
Lewis W. Heath
P. F. Shivers

George E. Tubbs

MARINE DIVISION

Joseph Van Blerck, *Chairman*

H. H. Brautigam
George F. Crouch
G. C. Davison
Kirk W. Dyer
H. E. Fromme
A. F. Milbrath
W. C. Morehead
Guy W. Vaughan

MISCELLANEOUS DIVISION

E. H. Ehrman, *Chairman*

F. C. Billings
Clarence Carson
W. C. Keys
W. R. Strickland
E. E. Sweet
Fred G. Whittington

MOTORCYCLE DIVISION

W. S. Harley, *Chairman*

George W. Dunham
C. B. Franklin
Major F. C. Hecox
F. W. Schwinn

NON-FERROUS METALS DIVISION

J. J. Aull, *Chairman*

E. Blough
G. K. Burgess
F. O. Clements
E. S. Fretz
H. L. Greene
Zay Jeffries
H. P. Parrock
W. C. Peterson

ROLLER CHAIN DIVISION

L. M. Wainwright, *Chairman*

Warren J. Belcher
John R. Cautley
William F. Cole
Herbert F. Funke

H. S. Pierce

SHAFT FITTINGS DIVISION

C. W. Spicer, *Chairman*

J. F. Barr
R. J. Burrows
W. D. Diefendorf
W. C. Lipe
William MacGlashan
M. W. H. Wilson

SILENT CHAIN DIVISION

A. Ludlow Clayden, *Chairman*

Warren J. Belcher
John R. Cautley
Herbert F. Funke
John C. Howe
E. W. Mills
F. L. Morse

H. S. Pierce

SPRINGS DIVISION

W. M. Newkirk, *Chairman*

B. H. Blair
E. A. DeWaters
S. P. Hess
Harry R. McMahon
R. A. Schaaf
F. A. Whitten

STATIONARY ENGINE DIVISION

H. N. Edens, *Chairman*

Theodore C. Menges
L. M. Ward

TIRE AND RIM DIVISION

S. P. Thacher, *Chairman*

Pneumatic Tires for Passenger Cars Sub-Division

S. P. Thacher, *Chairman*

W. H. Allen
E. G. Hulse
R. B. Naylor
J. C. Tuttle

W. S. Wolfe

Pneumatic Tires for Commercial Vehicles Sub-Division

W. S. Wolfe, *Chairman*

W. H. Allen
L. R. Davis
E. G. Hulse
R. B. Naylor
J. C. Tuttle

Pneumatic Tires for Airplanes Sub-Division

W. H. Allen, *Chairman*

R. B. Naylor
A. H. Petersen
S. M. Schott
J. C. Tuttle
W. S. Wolfe

Solid Tires Sub-Division

A. Hargraves, *Chairman*

W. H. Allen
L. R. Davis
Hugo Hoffstaedter
R. B. Naylor
A. H. Petersen

Pneumatic Tire Rims Sub-Division

C. C. Carlton, *Chairman*

W. H. Allen
E. K. Baker
W. N. Booth
Lewis Fine
J. E. Hale
J. W. Holt
S. M. Schott
J. G. Swain
J. H. Wagenhorst

Solid Tire Bands and Rims Sub-Division

W. N. Booth, *Chairman*

W. H. Allen
E. K. Baker
C. C. Carlton
L. R. Davis
Lewis Fine
J. E. Hale
A. Hargraves
J. W. Holt

TRACTOR DIVISION

E. A. Johnston, *Chairman*

H. C. Buffington
L. W. Chase
A. H. Gilbert
R. O. Hendrickson
Alfred Krieg
John Mainland
M. B. Morgan
Dent Parrett
C. B. Rose
A. W. Scarratt

TRANSMISSION DIVISION

A. W. Copland, *Chairman*

A. C. Bryan
L. C. Fuller
A. A. Gloetzner
W. C. Lipe
C. W. Spicer
W. G. Wall
E. E. Wemp
S. O. White

TRUCK DIVISION

L. P. Kalb, *Chairman*

F. G. Alborn
Ralph W. Austin
William M. Britton
A. K. Brumbaugh
E. L. Clark
J. R. Coleman
Francis W. Davis
Charles O. Guernsey
H. B. Knap
A. F. Masury
W. T. Norton, Jr.
A. J. Scaife
E. E. Wemp
F. A. Whitten
Capt. G. R. Young

Sub-Division on Wheels

E. E. Wemp, *Chairman*

R. J. Burrows
E. L. Clark
Herbert Jandus
George L. Lavery
W. T. Norton, Jr.
J. G. Swain
Herbert Vanderbeek
A. S. Van Haltern

MOTOR BOAT MEETING

AT the Motor Boat session of the Society held at New York City on the evening of Feb. 25, much of interest was presented in connection with the plans of the Society as to standardization work in the marine field, as well as several papers of decided value to those engaged in the design and production of motor boats and engines and parts therefor.

Joseph Van Blerck, chairman of the Marine Division of the Society's Standards Committee, presided and opened the meeting with a convincing statement as to the necessity of securing the support of the members in the standardization work which he explained is of great value and possibility to the motor boat industry. He argued, in addition, that the Society should hold frequent meetings at which motor boat engineering subjects would be taken up. There are encouraging signs that engineers and draftsmen working in the motor boat field will indicate in a concrete manner their appreciation of the benefits to be derived at joint engineering deliberations.

George F. Crouch, professor of naval architecture at Webb Naval Academy, discussed present-day requirements of marine gasoline engines as imposed by the design and service conditions of power boats. He stated that at least three types of engine are needed to meet the demands in this connection.

Commander Holbrook Gibson, of the Submarine Repair Base, League Island Navy Yard, Philadelphia, discussed extemporaneously the Diesel engine as installed in German submarines, speaking specifically of U117. After the signing of the armistice Commander Gibson inspected forty of the German submarines that were first surrendered and afterward the remaining portion of the German submarine fleet. He explained that practically the entire German submarine fleet was equipped with the four-cycle Diesel engine built by the Machinenfabrik Augsburg-Nürnburg in four sizes of 300, 550, 1300 and 1750 hp. respectively. All of the engines are six cylinder, except the 300-hp. installation which is ten cylinder. Commander Gibson commended highly the design and workmanship of the German submarine Diesel engine.

Mr. Van Blerck spoke on the possibility of quantity manufacture of marine engines adaptable for other purposes. He explained that the marine engine must be capable of running under full load for an indefinite time at an inclination of 12 to 15 deg. He told of a plan in the observance of which it was expected that the marine engine should have the advantages of quantity production. He exhibited views of a farm tractor engine, one of the features of which was a gear-driven fan. The flywheel of this engine which was designed for use in motor boats and motor trucks also is mounted in the rear. He announced that his experiments had indicated that it is not necessary to have both engines running outboard in a twin-screw boat; that is, that both engines can run in the same direction without objectionable operation. The advantage of such an arrangement is, of course, the possibility of using two "standard" engines in such construction, avoiding the employment of different directions of rotations of the oil-pump, magneto, water-pump, generator and starting pump, as well as the need of different camshafts.

Mr. Van Blerck also referred to the sliding reverse gear which had been incorporated in the powerplant described, this being an innovation in the marine field except in relatively few cases where automobile engines had been used in motor boats together with an automobile transmission. In this reverse gear installation the reverse speed is 82 per cent of the forward speed.

W. C. Davids presented, as collaborator with O. P. Sells, an abstract of papers presented at earlier meetings of the Society containing data on piston performance in engines.

Location of Flaws in Rifle-Barrel Steel by Magnetic Analysis[1]

By R. L. SANFORD *and* W. B. KOUWENHOVEN[2] *(Non-Members)* *Illustrated with* PHOTOGRAPHS *and* CHARTS

ONE of the practical applications of magnetic analysis consists of the detection of flaws in bar stock used in the manufacture of steel products. At the request of the Ordnance Department of the Army and the Winchester Repeating Arms Co., an investigation was undertaken during the war with the end in view of applying this method of magnetic analysis to the testing of rifle-barrel steel.

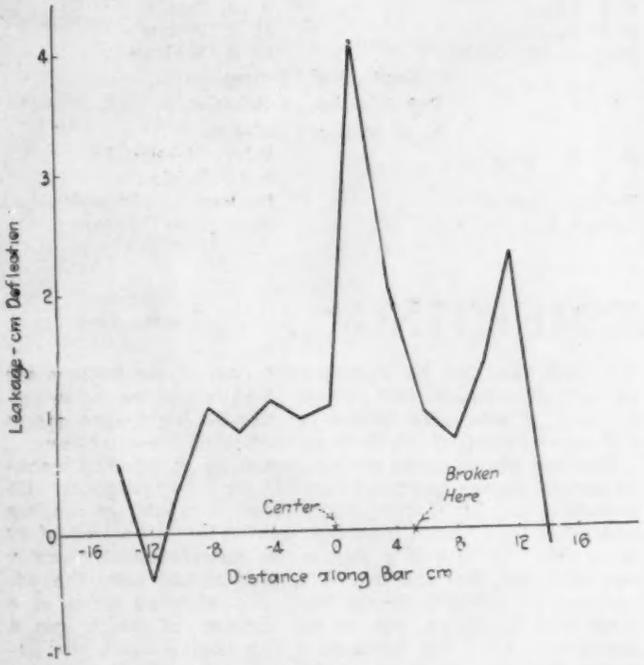


FIG. 1

In view of the fact that flaws, generally consisting of pipes or slag inclusions, interfere with the drilling of the barrels or may possibly affect their strength, it was considered that a non-destructive test which would detect and locate such flaws before further work had been done on the barrels would prove to be of great value. Such a method of inspection would make possible not only the rejection of faulty material, but also the acceptance of all the satisfactory bars in a given shipment and thus effect a great saving both of material and labor. It is the object of this paper to describe the apparatus used in the investigation and to present the results thus far obtained.

[1] Extracts from Scientific Paper No. 343 of the Bureau of Standards.

[2] The authors are associate physicist and consulting engineer respectively at the Bureau of Standards, Washington.

[3] Sanford, The Determination of the Degree of Uniformity of Bars for Magnetic Standards, Bureau of Standards Scientific Papers No. 295.

[4] Burrows, Correlation of the Magnetic and Mechanical Properties of Steel, Bureau of Standards Scientific Papers No. 272, p. 203.

THEORY

The method employed was that of the determination of the degree of magnetic uniformity along the length of the bars, based upon the theory that if a bar is uniform magnetically along its length, it is also uniform mechanically. A number of barrel forgings were first tested by a point-by-point method originally used for the examination of bars intended for magnetic standards and which has already been described.[3] Fig. 1 shows a specimen curve obtained by this method, and Fig. 2 is a photograph showing the flaw which was indicated by this curve.

Since this method is not adapted to the examination of very long bars, and is too time-consuming for a commercial test, it was decided to pursue a different method, which was substantially similar to that employed by Burrows[4] for the examination of steel rails. In this method the magnetizing force is applied by a solenoid, which surrounds the bar and travels along its length. Mounted within this magnetizing solenoid is a test coil by which variations in magnetic flux within the bar can be measured. If the bar is magnetically uniform along its length its permeability is constant for a given magnetizing force and the magnetic flux at each point as the solenoid is moving along is constant. If this is the case there will be no electromotive force induced in the test coil as the solenoid travels the length of the bar. If, on the other

FIG. 2

hand, the permeability is not constant, the flux will vary and a corresponding electromotive force will be induced in the test coil which, if the coils are moved at a constant speed along the bar, is proportional to the change in flux. If, instead of using a single test coil in the manner just described, we use two test coils connected in series opposition, we obtain a result that is practically not affected by slight variations in the magnetizing current during a run, as any variations in flux linked with one coil are neutralized by corresponding changes in the other.

Permanent records of the magnetic deflections are made by a photographic arrangement which consists of a long light-tight box upon one end of which is mounted an ordinary oscillograph drum that carries the photographic film. This drum is rotated at the proper speed by a belt connected to the driving motor of the apparatus. By contacts located at 1-ft. intervals on the driving cord, a light is flashed inside the box which makes a record on the film for each foot of travel, and thus affords

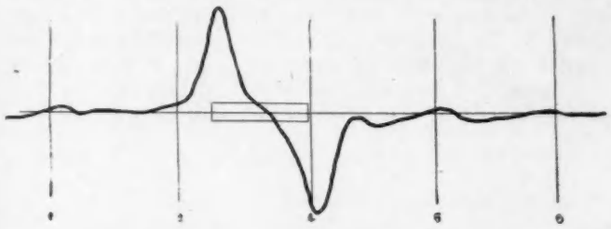


FIG 3

a means for locating the position on the bar of any observed non-uniformity.

When a photographic record indicating the magnetic uniformity of a bar is to be made, the procedure is as follows: The bar is clamped in the apparatus, the galvanometer circuit is then closed and the drum carrying the photographic film is given one complete revolution. The spot of light reflected from the galvanometer mirror thereby traces a straight line which serves as the reference axis. The switch is then closed and the magnetizing current is adjusted to the proper value by the regulating rheostat. With the galvanometer connected either to the single test coil or the differential coils as desired, the driving motor is then started and the coil is run the length of the specimen with the film holder rotating at a uniform speed. Most of the records have been made by running the coils in one direction with the galvanometer connected to the single test coil, and in the other direction with the galvanometer connected to the differential coils. Fig. 3 shows a record taken by the use of the single coil, and Fig. 4 shows the corresponding record taken with the differential coils. A rectangle is drawn on each of these records to show the position and extent

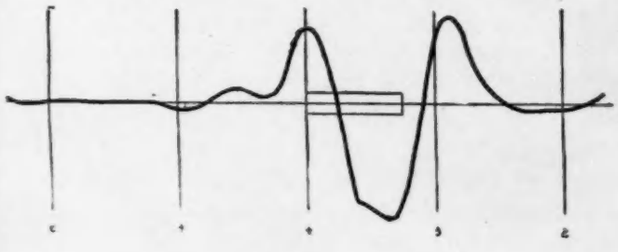


FIG. 4

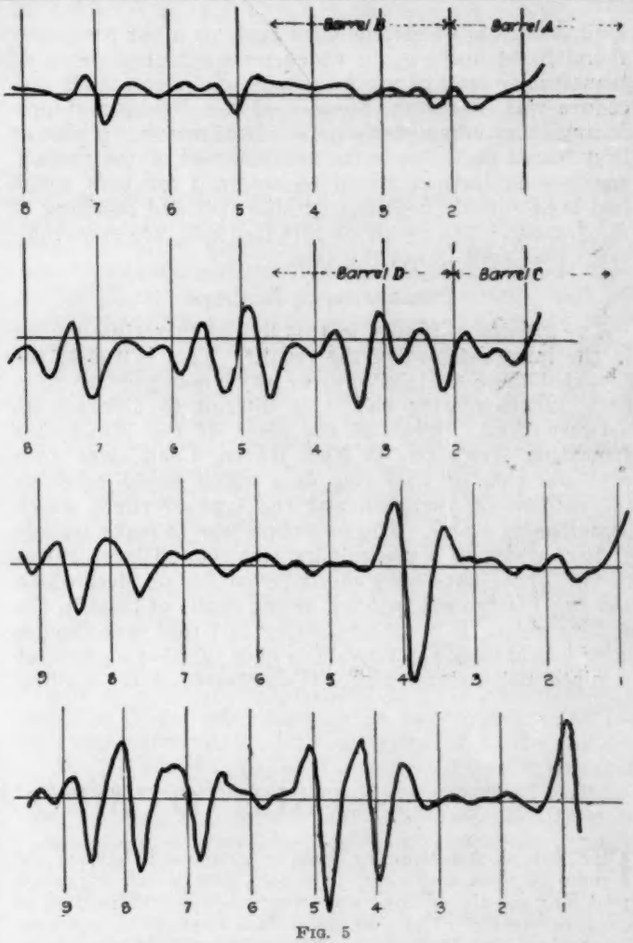


FIG. 5

of a strip of transformer iron which was attached to the bar to give the effect of a flaw.

PRELIMINARY STUDY AND ADJUSTMENT

After the apparatus was completed and set up in the laboratory, it was necessary to consider a number of points in connection with its operation and to decide upon the proper adjustment of the test coils. The points to be considered included the proper flux density, B, in the specimen, the proper speed for the moving coils, the period of the galvanometer and, as just mentioned, the best location of the test coils.

It was found that a flux density of approximately 15,000 gausses gives the best results. The speed of travel finally adopted was approximately ½ ft. per sec. It is necessary in order to insure that the record gives a true indication of the condition of the specimens that the galvanometer have a fairly short period. If the period is too long, the galvanometer does not follow closely the changes in the induced electromotive force. A period of approximately 1 sec. was found to be satisfactory.

A preliminary exploration to determine the flux distribution along the specimen for different positions of the magnetizing solenoid was made by a point-by-point method, using the single test coil connected to a ballistic galvanometer. Readings were taken upon reversal of the magnetizing current. The result of this exploration showed that, for a given magnetizing current, the flux is constant at different points along a uniform bar except for the regions very near the ends.

To study the effect of flaws varying in extent and

kind, a number of records were made on a bar previously found to be uniform, to which were attached strips of transformer steel of various shapes and sizes. This procedure was necessary, because of the difficulty of producing longitudinal flaws by artificial means. It was at first feared that, due to the sensitiveness of the method, spurious indications would be obtained for bars which had been slightly bent during shipment and handling at the factory. The result of this last test, however, indicates that such is not the case.

Experimental Results

The greatest difficulty in this line of investigation lies in the interpretation of the results. This is due to the fact that there are many causes which may produce magnetic inhomogeneity and it is difficult to differentiate between them. Work at the plant of the Winchester Repeating Arms Co. at New Haven, Conn., was done with the view of obtaining data which would establish the amount of variation and the type of curve which accompanies a pipe. The procedure was to make records of bars which in a preliminary test showed large variations. These bars were chosen from lots of steel which had previously been rejected as the result of tests in the drilling shop. It is an interesting fact that even though these lots of steel had previously been rejected on account of pipes, not a single pipe was discovered in the drilling tests on samples for which records of the magnetic uniformity had been obtained. This is true of all the steel examined up to March 31, 1919.

Figs. 5 and 6 show records of the degree of magnetic homogeneity of four bars of steel. These records were made with the differential test coils and with a fairly low sensitivity of the galvanometer. The portions of these bars from which barrel lengths were cut are indicated in the figure. Barrels *A*, *B*, *C* and *D*, cut from bars 1 and 9 and those cut from the entire length of bars 2 and 4 were sent to the shop for drilling tests. Barrels *B* and *D* gave trouble in drilling and each destroyed the edge of a drill, thus necessitating the use of a new drill to finish the bore. None of the other barrels gave trouble and the inside surfaces of all were smooth and bright. In view of the fact that additional data are necessary to draw satisfactory conclusions, the Winchester Repeating Arms Co. is continuing the investigation.

The results obtained demonstrated that the method is amply sensitive to detect and locate flaws. Further study is necessary to determine to what degree the sensitivity of the apparatus should be reduced in order not to cause the rejection of material which is satisfactory for all practical purposes, and also to determine the type and magnitude of the effect which will be produced by a pipe.

CAPE TO CAIRO AIR LINE OPENED

THE British Air Ministry recently announced that the air route between Cairo and Capetown, over which engineers spent long months of labor and overcame many difficulties, is ready for traffic. The time heretofore required to traverse the continent has been reduced from 10 or 12 weeks to approximately one, which means that secluded sections of Africa will now be opened up.

The total distance by the old methods of communication is 6223 miles for which from 59 to 75 days was required. The total flying distance of the aerial route, the Air Ministry points out, should not exceed 5200 miles. When the route is in regular operation, it is expected that an average flying speed of 100 m.p.h. can be maintained under favorable conditions. If this is possible, only 52 hr. actual flying time will be required to make the trip from practically one end of the continent to the other.

CHANGES IN THE ADMINISTRATIVE COMMITTEES

IN addition to the personnel of the various administrative committees of the Society which was printed in the January issue of THE JOURNAL, John Younger has been appointed a member of the Membership Committee. The complete list of members of this committee is W. A. Brush, chairman; A. C. Bergmann, H. A. Coffin, Dent Parrett and John Younger.

J. E. Maloney has been appointed a member of the Sections Committee. His associates are H. R. Corse, chairman; J. A. Anglada, A. D. Wilt, Jr., and C. B. Veal.

Design of Pneumatic-Tired Trucks

By C. M. McCreery[1] (*Non-Member*)

Chicago Truck and Tractor Meeting Paper *Illustrated with* Photographs and Drawings

THE truck designers are just now on the first flood of expansion in the world's next advance in transportation, but before discussing specifically the subject of this paper I would like briefly to point out how forms of transportation have extended civilization by broadening education and developing natural resources.

Going over the history of the rise of all nations briefly as a corollary to the transportation systems of those nations, we first know of little groups of barbaric individuals going about on foot; then of transportation by domestic animals, bringing forward Egypt, followed by Babylonia and Persia; then of Greece forging to the fore through transportation by hand-propelled boats. Larger boats pronounced Rome the leader and Rome also assured its permanent prominence by building good inland roads, making transportation by domestic animals faster and more efficient. Then followed sailing ships and we hear of Genoa and of Venice. Transportation over the ocean brought Spain and England to the front as leading world powers, and so they remained until the advent of the steam engine early in the nineteenth century.

Countries like America and Germany, with large areas of internal territory to develop, then began to assume the leadership and this was the status of affairs until our own day when the development of the internal-combustion engine and the pneumatic tire made possible the automobile, now an important form of transportation which will in the future take a more and more prominent place than any former means of transportation. What part the automobile plays from now on must affect vitally the civilization, progress and development of the whole world and its people, and how great its effect will be rests with you members of the Society of Automotive Engineers.

Possible Use of the Motor Truck

The field of motor transportation is beyond human comprehension. With billions of people to be transported every hour in the day and also with millions of tons of freight to be moved, the conveyance which completes the delivery with the least loss of time must predominate. Time is a factor of greater value than money, although we are all prone to determine costs by dollars rather than by the clock. So it is your task to produce motor trucks that will dominate as the world's greatest medium of transportation. It is our claim that the pneumatic tire, when its advantages are known and used to the fullest extent possible in truck design, makes possible this hope of future domination of the world's transportation by the motor truck.

Your past experience recalls the bicycle with first a steel, then a solid and then a pneumatic tire. People said the pneumatic tire would not do because it punctured, which was true; and that it cost more, which was true; but the bicycle did not continue on solid tires because the pneumatic tire gave it a wider field of usefulness. The rubber-tired carriage followed much in the same manner, except that the combination of the gas engine and pneumatic tire practically eliminated the horse from passenger transportation and bids fair to succeed the electric street car in that respect. Today there are more passengers per mile carried by automobile than by any form of rail or water transportation. But the carriage and transportation of merchandise has not developed as rapidly as has the transportation of passengers.

While motor trucks have already played an important part in world conditions, they have done so largely as auxiliaries to existing means of transportation. This we sincerely believe to be due to the continued use of solid tires and to the design of vehicles based on the use of solid truck tires. The lack of development of a suitable pneumatic truck tire by the tire designers is the most plausible reason for this condition, but during the past few years our development and research work have proved that pneumatic tires for trucks have been perfected to a point sufficient to justify a belief upon the part of the most conservative engineer that the pneumatic must be the ultimate tire for trucks in both passenger and freight transportation. Trucks designed to take the fullest advantage of the use of pneumatic tires can and will dominate the transportation of the world.

Most of you, I think, believe in the pneumatic tire principle for motor trucks, but are you convinced that they are practical and economical? To prove that they are one has but to search the records of experience of users. Any discussion of the relative merits of pneumatic versus solid tires must, of necessity, reduce itself to a comparison of the elasticity of compressed air with that of rubber. We all are accustomed to think of rubber as a very elastic substance, but it cannot be compared to compressed air in this respect. In one instance you ride on rubber; in the other, on air. Very little stretch of imagination is necessary to realize what would happen to a modern passenger car if it were to be equipped with solid tires. The reasons that motor trucks can be operated with any degree of success on solid tires are, that they are operated at relatively low speeds and are built very heavy so that they will endure the shocks and vibration to which they are subjected. If it is possible to use pneumatic tires on passenger cars to such advantage, it is reasonable to assume that certain advantages will result from their use on motor trucks.

It has been found that there are two fundamental advantages which result from the use of pneumatic truck tires: (*a*) increased cushioning, and (*b*) increased traction. The increased cushioning is the most important factor because it has a greater effect on the performance of the truck. Elsewhere I will show that the cushioning ability of a pneumatic tire is four times as great as that of a solid tire of the same carrying capacity. As a result of this greatly increased cushioning there are six distinct advantages gained from the use of pneumatic truck tires:

(1) Faster transportation
(2) Economy of operation
(3) Less depreciation of fragile load
(4) Easier riding
(5) Less depreciation of roads
(6) Lighter weight trucks

[1] Highway transportation division, development department, Goodyear Tire & Rubber Co., Apron, Ohio.

Each of these six points will now be discussed separately and, wherever possible, actual data will be presented to substantiate the claims made. The data has been obtained from truck operators, each of whom has taken two trucks of the same make and capacity, one equipped with pneumatic and the other with solid tires, and operated them under exactly similar conditions, over the same roads and carrying similar loads.

Transportation and Operation Economies

Faster transportation or quicker deliveries result from the increased cushioning of pneumatic truck tires. Operators have found this to be true because it is possible to obtain greater maximum and minimum speeds. Manufacturers of solid-tired trucks remove their guarantee if a speed of 11 or 12 m.p.h. is exceeded, while pneumatic-tired trucks are being successfully operated at 20 to 35 m.p.h. In ordinary city or farm hauling, however, we are more interested in a greater minimum speed than we are in a greater maximum. That is, in running over rough city streets or country roads, a solid-tired truck must operate at slow speeds because of the shock and vibration. It is therefore evident that if a truck on pneumatic tires will make more or longer deliveries in a given number of hours, its radius of operation is increased and also its earning power. Table I will show the increased mileage obtained with pneumatic tires by four truck operators

TABLE I

Operator	Truck Capacity, tons	Period, months	Mileage on Pneumatic Tires	Mileage on Solid Tires
A	2	6	6,414	4,476
B	3½	1	1,995	675
C	2	5	5,510	2,223
D	2	4	7,014	4,677

The economy of operating trucks on pneumatic tires has been shown by the experience of many users. There is a considerable saving in gasoline, oil and upkeep. Furthermore, depreciation charges can be reduced considerably. The saving in gasoline is due to a lower power consumption resulting from the increased cushioning obtained. It is a well-known fact that vibration in a machine of any kind results in a loss of power. The vibration caused by the solid tires is practically eliminated by the use of pneumatic tires. A solid tire, in rolling over an obstruction in the road, lifts the entire load on the tire four times as high as it would be lifted in the case of a pneumatic tire. Many roads are full of small obstructions so that a certain amount of power is lost in lifting the load of the truck over them. The relative cushioning ability of the two types of tire seems to indicate that the power loss from this cause is four times as much for solid as for the pneumatic type. There is also a saving in power in climbing hills. Because it is possible on pneumatic tires to approach a grade at a much higher rate of speed, less power will be consumed in climbing the grade; it will not be necessary to shift gears as soon, and possibly not at all. The saving in gasoline in the case of five truck operators is shown in Table II.

TABLE II

Operator	Truck Capacity, tons	Period, months	Miles per Gallon of Fuel: Pneumatic Tires	Miles per Gallon of Fuel: Solid Tires
A	2	6	5.77	3.98
B	3½	1	5.75	4.77
C	2	5	7.21	5.43
D	2	4	7.70	7.10
E	2	9	9.10	6.10

The saving in oil is probably due to the decreased vibration in all of the moving parts of the truck. The statistics from four truck operators, given in Table III, show that there is a considerable saving in oil consumption.

The upkeep or repair cost of a truck operated on pneumatic is much less than when operated on solid tires. This can be attributed to the decreased amount of vibration and the absence of severe shocks and jolts. It is found that parts do not have to be replaced and that the truck does not need to be overhauled as often. Reliable information has not been obtained, but the estimated saving in upkeep cost is from 25 to 50 per cent. There is also the subject of depreciation charges. As a result of experience our solid-tired trucks are depreciated on the basis of 60,000 miles of service, while the pneumatic-tired trucks are depreciated on a basis of 80,000 miles. It seems that 80,000 miles is too low, because there have been trucks on pneumatic tires which at the end of 250,000 miles were still in running condition. I believe that in the near future trucks will be depreciated on the basis of 100,000 miles. Taking all of these cost factors into consideration, it is found that the cost per mile of operating trucks on pneumatic tires is considerably less than that on solid tires. Referring again to the data obtained from truck operators who have kept accurate records, this claim is substantiated. (See Table III.)

Depreciation of Loads and Roads

Let us now consider the lessened depreciation of fragile loads resulting from the increased cushioning ability of pneumatic truck tires. This is considered by many users to be one of the most important advantages. For instance, in hauling fragile materials, such as bottled goods and eggs, there is very little, if any, breakage. The farmer experiences very little depreciation in the live stock and produce which he hauls to the markets.

There is also the easier riding made possible by the use of pneumatic tires. In the case of delivery trucks, the elimination of the vibration makes it possible for the truck driver and his helper to ride almost continually

TABLE III

Operator	Truck Capacity, tons	Period, months	Miles per Gallon of Oil: Pneumatic Tires	Miles per Gallon of Oil: Solid Tires	Cost per Mile in Cents: Pneumatic Tires	Cost per Mile in Cents: Solid Tires
A	2	6	104.0	59.0	45.0	56.3
B	3½	1	32.0	30.7	31.3	55.0
C	2	5	55.0	54.0	21.5	24.0
D	2	4	152.0	78.0	27.7	31.0

without fatigue. This is of vital importance, especially in long-distance hauling where it is necessary to drive for hours at a time. Easy riding is absolutely essential in passenger busses from the standpoints of comfort and speed.

Before the motor truck can be utilized to the full extent of its possibilities, good roads are essential. However, along with the legislation providing these, laws are being considered and in some instances passed restricting motor-truck traffic. A pneumatic-tired truck has very little more harmful effect on an improved road than a passenger car. This is not true of a solid-tired truck. Because of the vibration and sharp jolts common to solid tires, any road will eventually break down under such traffic. The effect on dirt roads is the most harmful. The narrow, hard tread of the solid tire will soon cut into the road, whereas the soft, wide tread of a pneumatic tire will not. Attempts have even been made to prohibit solid-tired trucks from using certain roads, and on some of the principal thoroughfares of our cities this has already been done. In such cases pneumatic truck tires have a distinct advantage.

The discussion so far has considered only increased cushioning. The advantages of increased traction will now be stated. Increased traction is made possible by the greater width of the pneumatic tires, their non-skid treads and their greater flexibility, which allows the surface of the tire to conform more nearly to the unevenness of the road, thereby getting a better grip. There is approximately twice the area of ground contact with pneumatic than with solid tires of similar capacities. As a result of this increased traction, we obtain reliability and safety. By reliability, we mean that it is possible for the truck to operate successfully over almost any kind or condition of road, and during all seasons of the year. By safety, it is meant that because of the increased traction of the tires, the truck will hold the road better and the brakes will be more effective.

We have so far discussed only the advantages of using pneumatic tires on trucks. I think it is advisable to mention some of the arguments which have been advanced against the use of pneumatics and truthfully answer them as experience has shown that they can be answered. All of the arguments against the use of pneumatic tires can be placed under either of the following: (*a*) cost and (*b*) practicability. The question of cost will be discussed first. It is now found that the matter of cost is subdivided into two parts: (*a*) initial cost, and (*b*) the possible loss due to injury and abuse. While it is true that the initial cost of pneumatic-tire truck equipment is greater than that of solid-tire equipment, it has been proved by the experience of many truck operators that this difference is more than offset by the greater earning power and the lower cost of operation. It has usually been found that in 4 to 6 months' time the increased cost of the pneumatic-tire equipment is completely wiped out. When the time comes that specially designed pneumatic-tired trucks make their appearance, this increased cost will be offset by the lower initial cost of the truck itself, and the greater economy of operation.

As to possible loss due to injury or abuse of the tires, it has been found that this is not a serious objection. There are innumerable instances where pneumatic truck tires have run from 12,000 to 20,000 miles on the original air. As the development of the tires themselves has progressed, so has the successful repairing of the tires been worked out. Repair molds and retreading equipment are now in use in many parts of the country and are being placed in other localities as rapidly as possible, so that it will be no more difficult to have a pneumatic truck tire repaired than any other part of the truck. While it is possible to injure pneumatic truck tires by abuse, this is no more likely to happen to the tires than to other parts of the truck, and it is possible to repair most of the injuries that tires receive from abuse. As truck operators become more accustomed to the use of pneumatic tires, such failures will be reduced.

Practicability of Pneumatic Tires

The practicability of pneumatic truck tires has probably been questioned more than has any other feature. The first thing to be discussed under this subject is that of delays due to changing tires. The amount of time necessary to change a tire depends in part upon the type of rim equipment. In the case of detachable rims, where it is necessary to remove the tire from the rim, replace it and then inflate it, not more than 30 min. is required to perform the entire operation. In the case of a demountable rim, a change can easily be made in 15 min. I am not trying to argue the relative merits of the two different types of rims, but is an occasional delay of from 15 to 30 min. to be considered when we know that we can save hours, not minutes, every day that the truck is in operation when it is equipped with pneumatic tires? The truck driver is not required to make a tire change very often.

It is often said that it is not possible to maintain the high inflation pressure required for pneumatic truck tires. Very little difficulty has been experienced in this respect. Most garages and service stations carry sufficient air pressure to inflate tires up to 42 by 9-in. size, and many are preparing to take care of the largest sizes. By the time the largest tires are in general use there will be generally available air at a sufficient pressure to keep them properly inflated. Trucks equipped with detachable rims, or operating in long-distance or inter-city service, are usually equipped with small air compressors. One objection which is not mentioned so much now as when pneumatic truck tires first made their appearance, is the danger of the high inflation pressures. Pneumatic tires are made to withstand three to four times the pressure carried in them.

The large outside diameters of the tires are often objected to because they affect the truck ability and because they raise the center of gravity. In changing over a solid-tired truck to pneumatics, there is the possibility of reducing the ability of the truck. Our experience has shown that unless the truck is operated over a very hilly route, its ability has not been noticeably affected. There have been too many successful change-overs to allow this to become a serious objection. Looking into the future, this question of truck ability and gear ratios will be taken care of by changes in design. So the question of change-overs is only temporary. Raising the center of gravity is not as serious as it might seem. The question of getting the load into the truck has not been a serious question at all. I do not believe that the loading platforms are of such uniform height that a little change in the height of the truck platform would make a great difference.

I have heard it said at different times that in case of a large rear tire suddenly going flat, the sudden drop of the truck might turn it over on a crowned road. When a big tire goes flat, it does not do so suddenly. The blow-out or cut in the tire is never very large, so that it takes some time for the air to escape. Therefore, the truck does not tip suddenly. Furthermore, when all of the air is out of a 48 by 12-in. tire, the largest now made, there

is a drop of only 5 in. Now assume that the tread of the rear wheel is 66 in., the list of the truck would be approximately 4½ deg., which is not enough to cause the truck to turn over, and to my knowledge no truck has ever overturned from this cause. The sizes that have been worked out during the past 6 years are now standard. Table IV gives the rim sizes, normal tire sizes and the tires which can be fitted as oversizes.

TABLE IV

Rim Sizes, in.	Normal Tire for Rim, in. (Original equipment on new trucks)	Oversize Tire for Rim, in. (Intended for convenience of consumer and not for original equipment)
34x 5	34x 5	36x6
36x 6	36x 6	38x7
38x 7	38x 7	40x8
40x 8	40x 8	42x9
[3]40x 8	42x 9	
[3]44x10	[3]44x10	
[3]48x12	[3]48x12	

[3]These sizes are not S. A. E. tire and rim sizes.

You will notice from this table that there is no oversize possibility when 9, 10 and 12-in. sizes go out on new trucks because the oversizing plan falls down above the 9-in. size, on account of the size and stiffness we are forced to build into the beads as designed at present.

I would like to discuss the questions regarding loads and inflations at considerable length but can only do so briefly and tell something of what pneumatic tires can stand in the way of flexing, so that the table of loads and inflations can be understood. (See Fig. 1.)

FIG. 1

FIG. 2

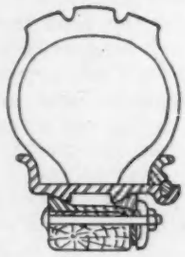


FIG. 3

To the best of my knowledge tires give the greatest satisfaction when they have plenty of cushioning and not too much flexing. The latter condition is what breaks down a tire, when run under conditions which produce a deflection of from 12 to 15 per cent of the section diameter, or 12 to 15 per cent of the height above the rim. The deflection can be controlled by regulating either the load or the pressure, or both. Table V gives the standard maximum load and required inflation. The pressures are practical to maintain, the tires are built accordingly and we get satisfactory practical results in first cost and mileage delivered, if the table is followed.

TABLE V

Tire Size, in.	Extreme Maximum Allowable Load per Tire, lb.	Inflation Pressure, lb. per sq. in.
34x 5	1,700	80
36x 6	2,200	90
38x 7	3,000	100
40x 8	4,000	110
42x 9	[5]5,000	[3]120
[3]44x10	[5]6,000	[3]130
[4]48x12	[4]8,500	[4]140

[3]S. A. E. Recommended Practice only.
[4]Not S. A. E. Standard as yet.
[5]This report was submitted to the Standards Committee as general information only. It was not accepted by the Society for publication in the S. A. E. Handbook.

I want to take this opportunity to mention under-inflation and overload. Both evils result in an excessive deflection of the tire which means that an excessive shearing action is put upon the rubber between the plies and also upon the cushion built into the tire between the tread and the plies. This in turn results in a separation of the parts and the tire is on the road to failure.

Next I will show how the loads recommended for the tires are adapted to the various sizes of trucks. The Tire and Rim Division of the S. A. E. Standards Committee investigated this, and its findings appear on page 24[5] of the Report of Divisions of Standards Committee, dated June 23, 1919. The weights of practically all makes of trucks were ascertained and plotted, and it was found that all trucks of a given capacity weighed nearly the same when under full load, and it was possible to specify what the pneumatic tire equipment should be on any truck of a certain carrying capacity. The bodies of all the trucks in question were assumed to be standard stake bodies so, of course, heavy bodies might in some cases necessitate a revision of the committee's findings. Table VI is similar to that submitted as general information to, but not accepted by, the Standards Committee. The changes are in the tire equipment recommended for front wheels on trucks of 1½ tons and under.

TABLE VI

Size of Truck, tons	Front Wheels: Maximum Weight per Wheel, lb.	Front Wheels: Pneumatic Tire Size, in.	Rear Wheels: Maximum Weight per Wheel, lb.	Rear Wheels: Pneumatic Tire Size, in.
¾	800	4, 4½ or 5	1,600	34x 5
1	1,000	4½ or 5	2,100	36x 6
1½	1,200	4½ or 5	3,000	38x 7
2	1,500	34x5	3,500	40x 8
2½	1,800	36x6	4,000	40x 8
3	2,000	36x6	5,200	44x10
3½	2,100	36x6	5,700	44x10
4	2,300	38x7	6,500	48x12
5	2,700	38x7	7,800	48x12

If you check up any make of truck loaded to its rated capacity, you will be surprised to find how closely the weights on the tires come to the figures in the table.

Wheels, Rims and Valves

There are now in satisfactory service wood wheels, 8 or 10 designs of cast-steel wheels and disc wheels. On all of these can be mounted either the so-called demountable type of rim or the so-called detachable rim. The demountable rim means one on which is carried an inflated tire. The detachable rim is perhaps a misnomer, because the rim is not detachable but the tire is removed by detaching a side ring from the rim, on which spare tires cannot be carried ready inflated. (See Figs. 2 and 3.)

The arguments for the demountable rim are the inflated spare tire and the short time required for a tire change. Against the demountable are the extra weight required, the extra cost and the possibility of mechanical troubles. The detachable rim arguments are the exact opposite. In its favor are less weight and cost and a greater freedom from mechanical troubles. Against it are the arguments of time required for a tire change and means of inflating the tire after it is changed.

Tubes for pneumatic truck tires must be designed and compounded so as to retain as much of their original

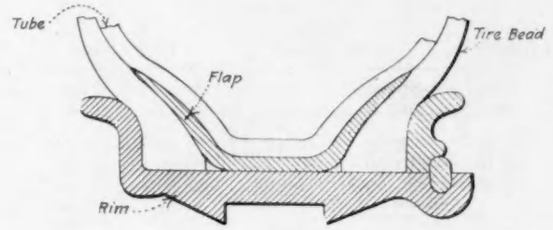


FIG. 4

strength and shape as possible, after being subjected in service to more or less heat and to continued flexing. Tubes are on a par with the casings in development and render satisfactory service even in the largest tires. (See Fig. 4.)

Flaps assume considerable importance in tires inflated to recommended pressures. It is important that the flap fit well so that there will be no adjustment when the tire is inflated, causing a localized stretch in the tube at the edge of the flap.

The valve question had to be approached from two angles; first, from the standpoint of holding air at pressures from 90 to 140 lb. and second, from the standpoint of ease of tire change. The valve-insides on all tubes 6 in. and larger are of a heavy-duty type, different from the ordinary valve-insides in construction, but the two are interchangeable in any valve stem (See Fig. 5). On the 10 and 12-in. sizes, which are inflated to 130 and 140 lb. per sq. in. respectively, even the heavy-duty type is at present unsatisfactory when used alone, so a combination is used in the form of a heavy-duty valve-inside and a needle valve operated by a hand-screw.

Another important item is the arrangement of tires on the trucks. There are three possibilities: (*a*) the conventional truck with single pneumatic tires on four wheels, (*b*) dual pneumatics on the rear, and (*c*) trucks with six or more wheels. The company with which I am connected is primarily interested in carrying trucks on pneumatic tires, so it has tried the three possibilities. As a result it looks unfavorably only on the dual pneumatics, two tires on the same wheel. The dual idea was tried and abandoned several years ago, but it crops up now and I want to set the reasons for dropping it clearly before you. The dual tires do not share the load equally, because the inflation is seldom kept alike in both tires and because on crowned roads, and more particularly rough roads, one tire takes more than its share of the load. Dual tires are too easily abused and prove more expensive than either of the other two possibilities.

The company came to realize some time ago that the introduction of the pneumatic tire for motor trucks would have a material bearing upon the design of the truck itself to get the most good from the use of such a tire. For this reason we have been making a study of the problem, and have at this time certain considerations to present.

Factors Influencing Truck Design

The main factors bearing upon the problem of truck design for pneumatic tires are as follows:

(1) Speed, including road and engine speeds, rear axle gear reduction and airbrakes
(2) Traction, including engine torque and transmission gear reductions
(3) Shock effects, including stresses introduced and the necessary factor of safety of sprung and unsprung parts
(4) Emergency equipment, including tire pumps and spare tires

Table VII shows road speeds that I consider satisfactory, together with the usual rear-tire specifications for various sizes of truck. The engine speeds are figured on the basis of 1200 ft. per min. piston speed, which value I believe can be considered a good average. However, some engines on the market may not operate successfully at this speed and again others can stand a higher speed. Higher speeds set up considerable vibration and add discomfort to driving. In general it may be noted that solid-

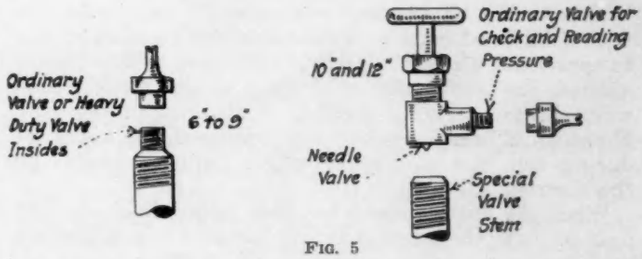

Ordinary Valve for Check and Reading Pressure
10" and 12"
Ordinary Valve or Heavy Duty Valve Insides
6" to 9"
Needle Valve
Special Valve Stem

FIG. 5

tire gear-reductions range from 7 to 16, whereas recommended ratios for pneumatic tires range from about 5 to 8.

Table VIII shows a study in computed speeds over a given course which corresponds somewhat to the course from Akron to Cleveland, going by the way of Tallmadge in the one case, and by way of Akron's North Hill in the other. K-1 represents a "standard" make of truck in our transportation service. It will be noted that a normal

TABLE VII

Present Solid Tire Gear Ratios	Solid Tire, Average Governed Speed, m.p.h.	Truck Capacity, tons	Recommended Speed for Pneumatic-Tired Truck, m.p.h.	Rear Tire Size, in.	Speeds: Rear Wheel, r.p.m.	Speeds: Engine, r.p.m.	Pneumatic-Tired Truck Rear Axle Gear Reduction
7	17	1	30	36	280.0	1,450	5 18
8	17	1½	30	38	265.4	1,450	5.47
9	15	2	30	40	252.1	1,325	5.52
10	15	2½	30	42	240.1	1,325	5.26
11 to 12	13	3½	25	44	191.0		6.28
				[a]38	221.1	1,200	5.43
12 to 13	11	5	25	48	175.0		6.86
				[a]40	210.1	1,200	5.72
14 to 16	9	7	20	[a]42	160.1		7.50
				[a]44	152.8	1,200	7.85.

[a]Six-wheel combination.

TABLE VIII

	Solid Tire, 1260 r.p.m.		New Job, 1400 r.p.m.		Six-Wheel, 1600 r.p.m.		K-1, 1400 r.p.m.	
	Speed, m.p.h.	Time, hr.	Speed, m.p.h.	Time, hr.	Speed, m.p.h.	Time, hr.	Speed, m.p.h.	Time, hr.
TALLMADGE								
3 miles, 6 per cent grade	6.66	0.45	8.57	0.35	6.25	0.48	8.15	0.37
3 miles, 3 per cent grade	11.00	0.27	17.90	0.16	10.50	0.29	14.10	0.21
34 miles, level	11.00	3.09	25.00	1.36	25.00	1.36	23.00	1.48
Total time	.....	3.81	.....	1.87	.....	2.13	.....	2.06
Average speed	10.50	....	21.40	....	18.70	....	19.40	....
NORTH HILL								
1 mile, 12 per cent grade	3.52	0.28	4.63	0.22	3.47	0.29	3.78	0.26
2 miles, 6 per cent grade	6.66	0.30	8.57	0.23	6.25	0.32	8.15	0.25
3 miles, 3 per cent grade	11.00	0.27	17.90	0.16	10.50	0.29	14.10	0.22
34 miles, level	11.00	3.09	25.00	1.36	25.00	1.36	23.00	1.48
Total time	.....	3.94	.....	1.97	.....	2.26	.....	2.21
Average speed	10.10	....	20.30	....	17.70	....	18.10	....

speed of 25 m.p.h. is required to double the average speed of a solid-tired truck whose normal or governed speed is 11 m.p.h. The reason, of course, is that the solid-tired truck has a higher tractive ability in high gear, and hence is able to maintain its normal speed over a great many grades.

Fig. 6 shows how time can be conserved by a careful study of gear reductions. The saving on a course 4 miles long with grades as shown amounts to 20 per cent.

On account of the higher speed of the pneumatic-equipped trucks, it is necessary to equip them with brakes having 100 per cent more capacity than is the case with solid-tired trucks. This raises the question of how to apply such a capacity easily. In answer to this I would suggest the application of brakes to the front wheels when all the capacity possible is attained in the rear. Front-wheel brakes are now well developed, due to the use during the war of trailers which required brakes on the steering axles.

When the normal speed has been determined, we may next consider the tractive ability required to get over the road without the inconvenience of shifting gears too often. The tractive factors that I consider desirable and satisfactory are shown in Table IX, figured according to the formula at the head of the table.

The engine torque required to give these tractive factors is also shown together with the sizes of engines on the market today that develop the torque required. There are conditions however where larger engines may be desirable. I believe smaller engines will not give satisfactory speed. The striking difference between tractive factors for pneumatic-equipped trucks and solid-tired trucks can be accounted for by the fact that the former, when approaching a hill, has about four times the momentum of the latter and will carry itself over a large percentage of hills without increased power. However, with the increased power applied it will ascend a very long steep grade before requiring a shift in gears.

The figures here recommended are based upon numerous trials of trucks of various capacities in our transportation. With the high-gear tractive ability and engine size determined, we have next the low-gear ability to consider. We find a tendency toward a low-gear ratio in the transmission of 6 to 1 in 5-ton trucks. This, in connection with present solid-tire axle-ratios gives a tractive factor of 0.42. A desirable low-gear ability for trucks equipped with pneumatic tires may be given as 0.50 although I do not feel that it should be less than 0.30.

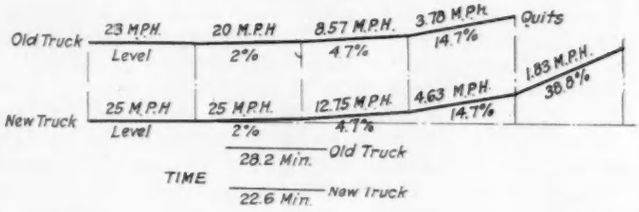


FIG. 6

Table X shows the low-gear transmission ratios necessary to give these tractive factors. It will be observed that these ratios are considerably different from present practice. The question naturally arises, how can this best be handled?

Fig. 7 shows a conventional design of transmission to give about 14 to 1 reduction in low gear. Superimposed

TABLE IX

Tractive Factor =
Maximum Engine Torque in pound-inches × Efficiency of Transmission (0.90 in high gear, 0.85 in other) × Gear Reduction ÷ Weight of truck and load in pounds × ½ the diameter of rear tire in inches

Truck Capacity, tons	Pneumatic Tire Tractive Factor	Corresponding Solid Tire Tractive Factor	Engine Torque, lb.-in.	Engine Sizes on Market that Develop Approximate Torque Required, in.		
1	0.07	0.085	1,650	3¾x5	4 x5	3¾x5⅛
1½	0.06	0.085	1,950	4 x5½	4⅛x5¼	4⅛x5½
2	0.06	0.083	2,500	4½x6	4½x6¼	4½x5½
2½	0.06	0.082	3,000	4¾x6	4½x6¼	
3½	0.05	0.077	3,000	4¾x6	4½x6¼	
5	0.04	0.070	3,200	5 x6	4¾x5½ (6 cyl.)	
7	0.04	0.060	3,300	5 x6	4¾x5½ (6 cyl.)	

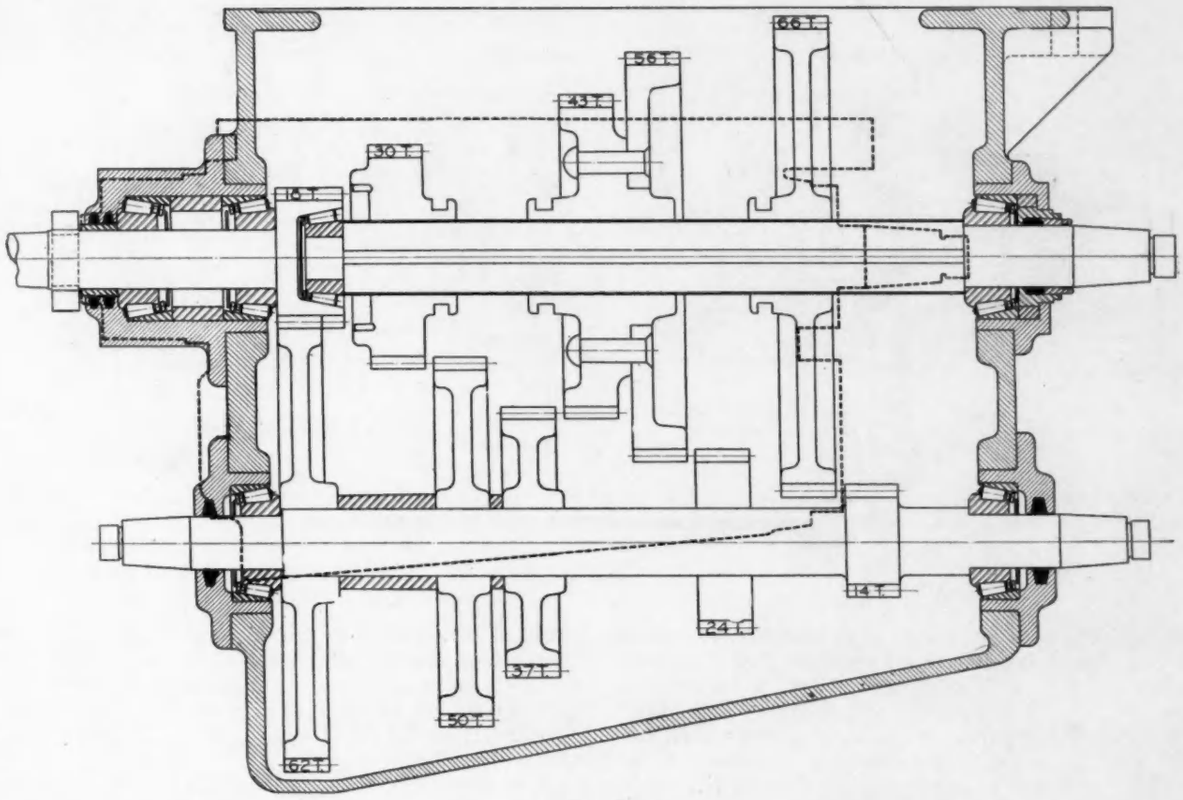


FIG. 7

upon this, in dotted lines, is a regular transmission of about 5 to 1 low-gear reduction, in general use today. By examining this, it becomes evident that such a construction cannot well be considered. The next best and easiest arrangement is as shown in Fig. 8 where we have a three-speed unit transmission of regular design in combination with an auxiliary transmission having 3½ to 1

TABLE X.—TRANSMISSION LOW GEAR REDUCTION

Truck Capacity, tons	Low Gear Tractive Factor	Engine Torque, lb.-in.	Total Gear Reduction	Rear Axle Reduction	Transmission Reduction in Low Gear	Trans. Dedn. 0.30 Tractive Factor
1	0.50	1,650	49.1	5.18	9.47	5.68
1½	0.50	1,950	51.0	5.47	9.32	5.58
2	0.50	2,500	49.5	5.26	9.42	5.64
2½	0.50	3,000	51.7	5.52	9.37	5.62
3½	0.50	3,000	61.4	5.43	11.30	6.77
5	0.50	3,200	79.7	5.72	14.00	8.40
7	0.50	3,300	105.0	7.50	14.00	8.40

reduction which, combined with the 4 to 1 low-gear reduction in the unit set, gives 14 to 1 total transmission low-gear reduction. Dotted lines in the figure show the unit and auxiliary, whereas the light full lines represent the standard amidships construction. Fig. 9 shows a more compact gear-set system that involves the combination into one of the two sets above mentioned.

We will next take up the consideration of allowable stresses. Any fixing of allowable stress requires an investigation of the cushioning effect of pneumatic, as compared with solid, tires.

Fig. 10 shows the rate of deflection of the pneumatic tires and their corresponding solid tires, together with a curve showing how the solid tire depreciates in resilience with age and wear. It will be seen here that for a given load the pneumatic tire deflects four times as much as a solid tire.

On this basis, Fig. 11 indicates that the stresses in unsprung parts due to shock, build up to the same amount whether pneumatic or solid tires are used. There is,

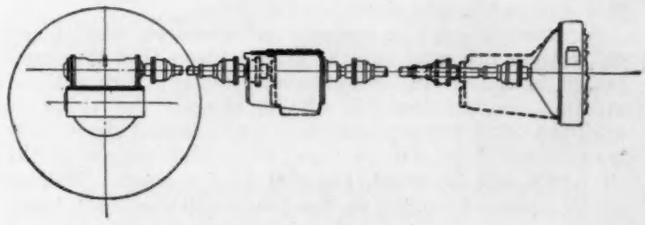


FIG. 8

however, a difference in the time element; that is, the time required to build up the stress due to shock is twice that of the solid tire, in the case of pneumatic tires. If any reduction of the factor of safety is justifiable, it appears that it is to be done on this basis. As yet we have been unable to conceive a method of testing or experimenting that will prove conclusively how much the factor of safety can be changed because of this time element. Any suggestions along this line will be appreciated. I have, however, taken a 2-ton chassis and installed engines designed for 3-ton trucks, and made the

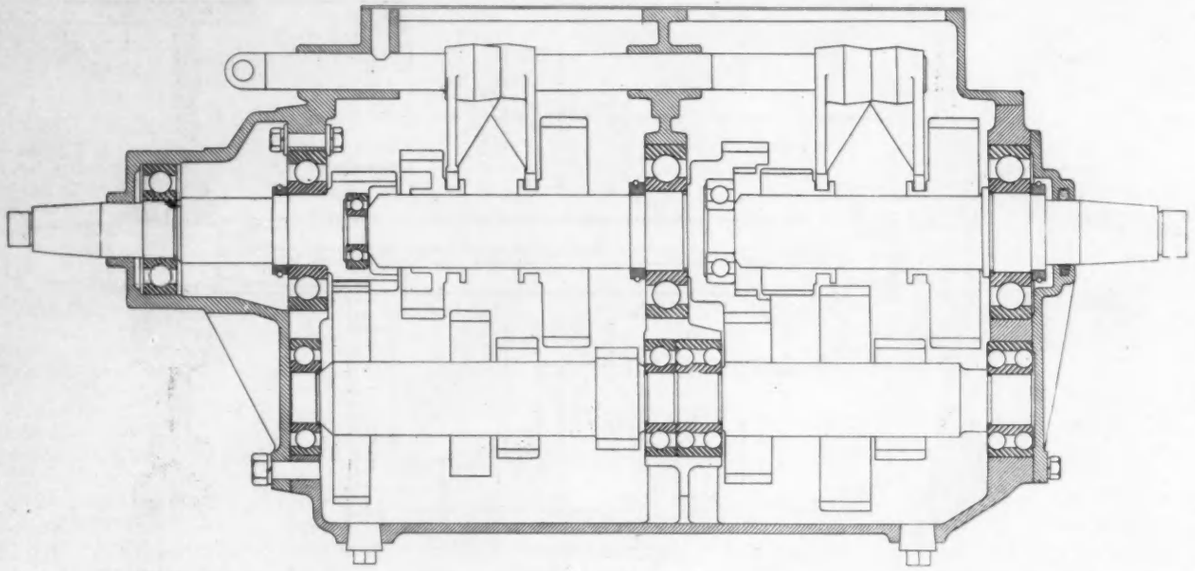


FIG. 9

trucks haul 3½ tons of freight very successfully. From numerous other trials like this I conclude that if a factor of safety of 6 is safe with solid tires, a factor of 5 can be allowed for pneumatic equipment. This refers mainly to unsprung parts. Regarding sprung parts, it is difficult even to estimate the allowable stresses. It appears that the factor of safety can be reduced but not so much as in the case of unsprung parts. For instance, with pneumatic tires, the frame is not deteriorated by shocks in the same manner as when solid tires are used, the rivets stay tight, etc. Fig. 12 shows how the engine is cushioned by the pneumatic tire.

EMERGENCY EQUIPMENT

The method of carrying spare tires is quite a problem. However, it is worth while to make provision for carrying this spare equipment when designing the body or chassis. Probably the best solution is to place a compartment directly back of the driver's seat so that the tire can be removed and replaced without disturbing the pay load. The next best place is under the chassis frame at the rear and at the side opposite the muffler.

A power-driven tire pump is indispensable when pneumatic tires are used for the main reason that the truck cannot be moved any distance on a deflated tire without causing serious damage. Even though the truck is equipped with demountable rims and inflated spare tires are carried, there will be times when a single use of the air pump will be worth the cost of the pump. Having the tire pump installed on the truck will facilitate keeping the tires properly inflated and produce better tire mileage; this factor alone will easily overcome the cost of the tire pump and in addition yield good returns. This pump should be designed to drive from the transmission. At least this is preferable; a pump mounted on the engine adds to the congestion under the hood, and the small shaft available, already carrying the water pump and magneto, is not strong enough for an air pump. The pump can best be air-cooled, due to its intermittent service and the difficulty of embodying any efficient water-cooling arrangement without much expense. Also, its lubrication should be well worked out, with a view toward avoiding any oil discharge that would injure the inner tube. The speed at which the pump is to operate should not exceed one-half of the governed engine speed and in no case should it be over 600 r.p.m. The pump should operate to permit complete inflation of a 10-in. tire in approximately 10 min. This figure is being realized by several pumps now on the market.

It is worthy of note that a 44 by 10-in. tire contains 5700 cu. in. of air space. The area of the orifice through which the air must pass in a standard tire valve is 0.00307 sq. in. Hence, if a pressure of 200 lb. per sq. in. upon this orifice is assumed, the minimum time of inflation of a 44 by 10-in. tire would be about 6 min.

It is debatable whether a two-stage pump is necessary in this service. Certainly, present design would not indicate this to be true, as there are single-stage pumps quite as efficient as any two-stage, for this relatively small volume. It is suggested that the air intake on the pumps be piped to a clean point on the truck, perhaps under the seat, thus preventing dirt from being sucked into working parts. At least 6 ft. of copper tubing

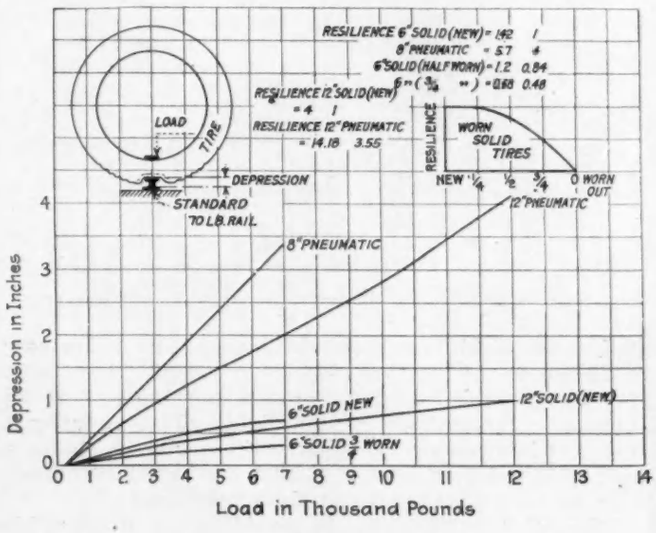
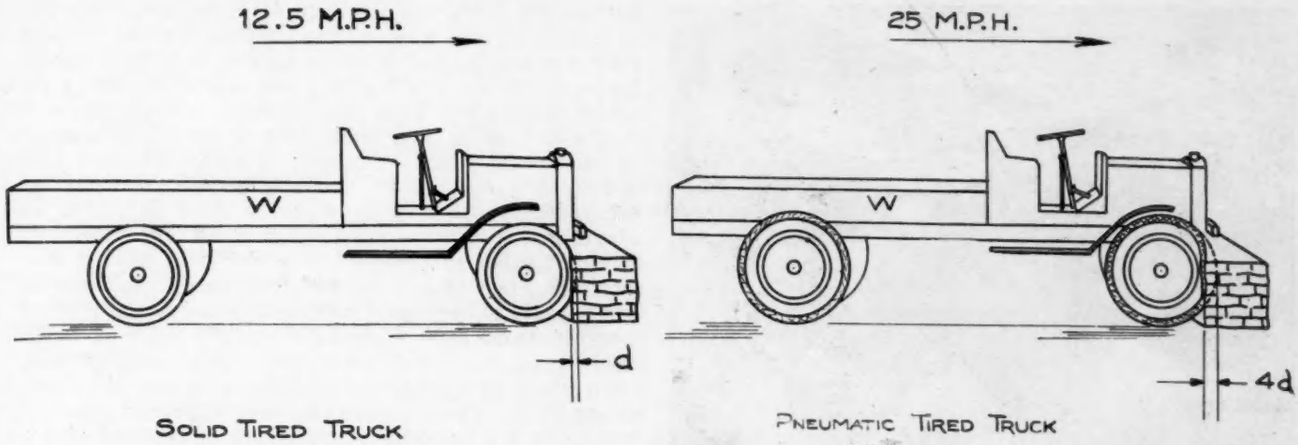


FIG. 10



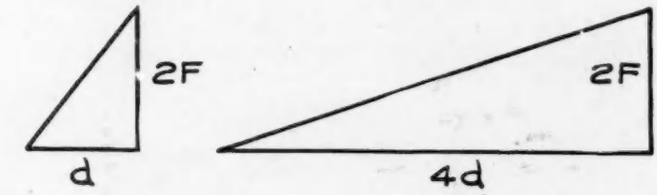
SOLID TIRED TRUCK

PNEUMATIC TIRED TRUCK

$$E = \frac{WV^2}{64.4}$$

$$E_s = \frac{20{,}000 \times 18.3 \times 18.3}{64.4} = \frac{6{,}700{,}000}{64.4} = 104{,}000 \text{ Ft.-Lb.}$$

$$E_p = \frac{20{,}000 \times 36.6 \times 36.6}{64.4} = \frac{26{,}800{,}000}{64.4} = 419{,}000 \text{ Ft.-Lb.}$$

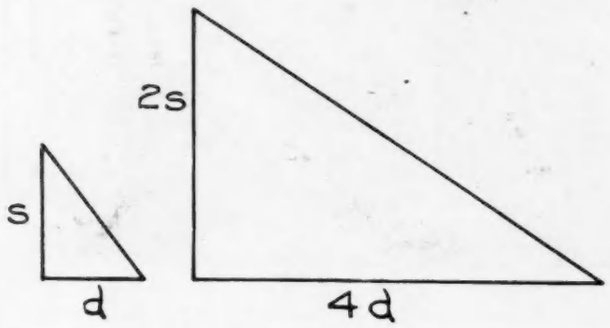


F_d = Work done in stopping truck
$= F_s = F_p$

$$F_s = \frac{104{,}000}{d}$$

$$F_p = \frac{419{,}000}{d}$$

$$\frac{104{,}000}{d} = \frac{419{,}000}{d}$$



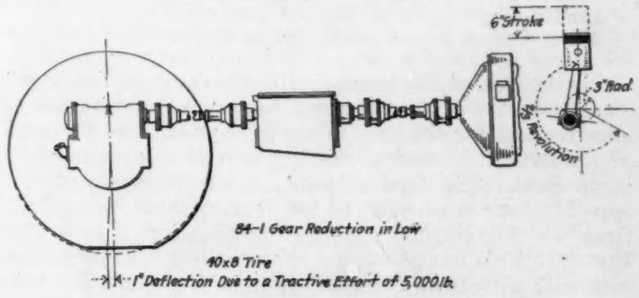
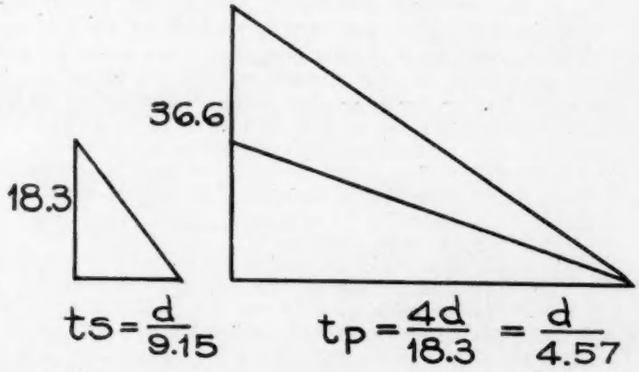


FIG. 12



$$ts = \frac{1}{2} tp$$

FIG. 11

$n = \frac{t}{c}$, and

$t = \frac{d}{s} \times g \times c$, where

n = Revolutions of engine due to circumferential deflection of tire
t = Travel of crankpin
c = Circumference of crankpin circle
d = Circumferential deflection of tire
s = Circumference of tire
g = Total gear reduction

$$t = \frac{1}{40 \times 3.1416} \times 84\ (6 \times 3.1416) = 12.61$$

$$n = \frac{12.61}{18.85} = 0.67$$

should be placed between the pump and the hose, to avoid burning off the latter. The advisability of placing a small receiving chamber in the line has been suggested. This would equalize the air pressure and assist in reducing the excessive temperature of the air delivered.

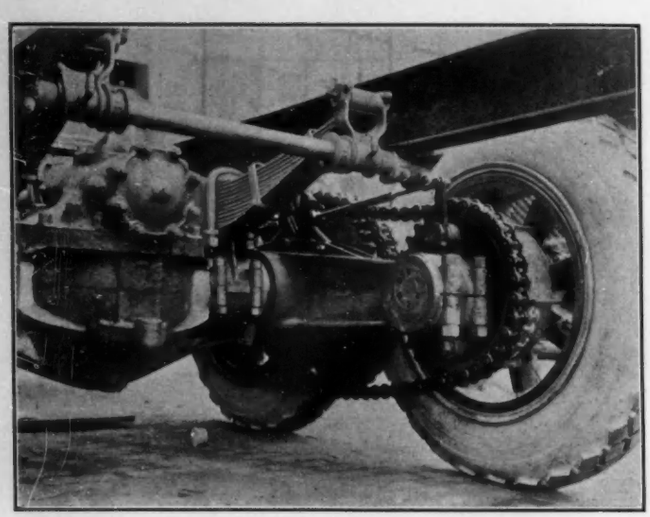


FIG. 13

NEW SIX-WHEEL TRUCK

It may interest you to know that my company has under construction at present two 5-ton trucks of six-wheel design. These trucks will have the following specifications:

Engine, 5 by 6 in.; four-cylinder
Transmission, unit power plant and auxiliary; 14 to 1 low-gear reduction
Rear axle, Goodyear tandem rear. Made up of two worm-drive axles, intended for use on 1½-ton solid-tire trucks in one case, and two internal-gear axles of the same rated size in the other. The rear-axle gear-reduction is 5.8 to 1
Front axle, intended for a 3½-ton solid-tire truck
Frame, 3 by 7 by ¼ in., pressed steel
Tires, 40 by 8 in. pneumatic; six in all

On account of the large size and weight of the 48 by 12-in. pneumatic tire for the 5-ton trucks, it was necessary to consider the application of four tires to the rear of the truck, instead of having two of the excessively large ones. The first attempt at an arrangement for applying four small tires to the rear, without using dual tires, which is considered out of the question, is shown in Fig. 13. This consisted of a more or less standard rear axle with a walking-beam adapted to each end and with the wheels mounted upon trunnions from this walking-beam, the springs being mounted upon the axle and attached to the frame on the inside. A chain drive, which, by the way, is about the only feasible drive with this arrangement, was used. This construction ran successfully for about 10,000 miles before a serious failure occurred. Some inconvenience was caused by having the chains jump off and inability to get a brake mechanism that would work. The main point against this design is its enormous weight; however, it served to show that satisfactory tire mileage could be secured from such an arrangement and that there was a good possibility of adapting four relatively small tires to the rear wheel. To develop further this point, the tandem-axle construction, shown in Figs. 14 and 15, was built up. This construction appears to have good possibilities and has at present operated about 3300 miles, 1000 to 1200 miles of which has been on an entirely rough and uneven country road, so rough in fact, that it was difficult to keep the front spring tight. The principle of construction can easily be seen by referring to the illustration. It consists of two small live-axles, with inverted springs pivoted at the center upon the frame and with the ends resting upon the axles. The axles are tied together by crossed torque-rods. The drive is passed through the forward to the rear axle by a short propeller-shaft.

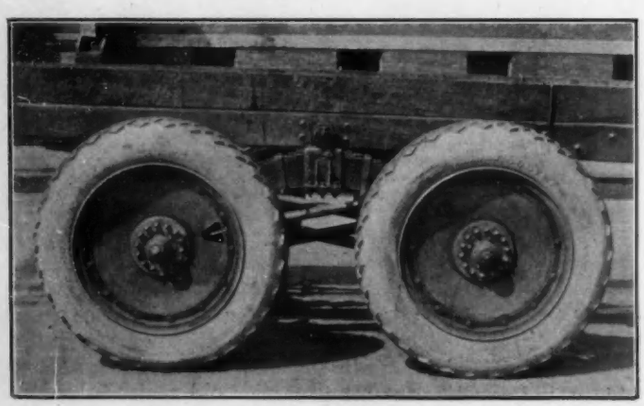


FIG. 14

Fig. 16 indicates the possibility of using four springs to tie the axles together and is a very feasible construc-

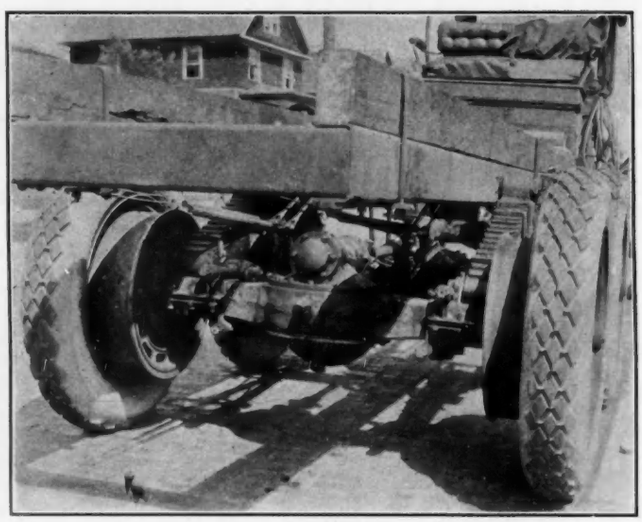


FIG. 15

tion to use in connection with the tandem rear-drive. While the drive described here is worked out for a truck of 5-tons' capacity, it is equally applicable to one of 7-tons' capacity. Some of the advantages the six-wheel truck has over the regular type of the same capacity, on 48 by 12-in. pneumatic tires are:

(1) Saving in cost
(2) Reduction in weight
(3) Reduction in axle cost
(4) Increased traction
(5) Better riding qualities
(6) Less damage to roads
(7) Larger brakes
(8) Greater operating radius

Compared to the pneumatic-tired four-wheel-truck using tires of large diameter, namely 48 by 12 in., the saving by using four smaller tires is about $500; sufficient to purchase three or four complete spares.

Each 40 by 8-in. tire weighs only 119 lb., whereas each

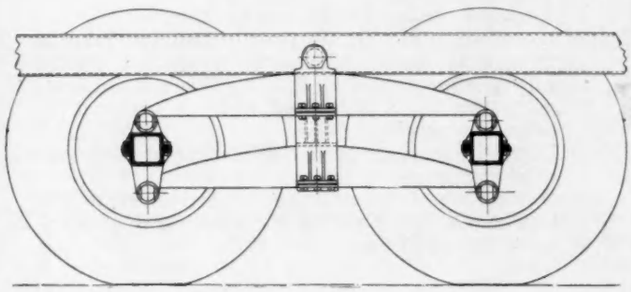


FIG. 16

48 by 12-in. tire weighs 398 lb. Carrying a spare tire in each case, the total weight reduction in tires alone amounts to over 500 lb. in favor of the 8-in. size. Then again, the 8-in. spare can also be used as a front-wheel spare; this reduces the tire investment still further.

The use of two rear axles in tandem, results in the employment of small axles which are in large production with consequent lower costs, instead of the large sizes that are made only in small quantities with relatively high costs. The actual saving amounts to about $120 per truck. This means that the tandem rear-drive mechanism costs somewhat less than standard two-wheel rear-drive for heavy trucks.

While four 8-in. wheels with the brake-drums, etc., weigh 77 lb. more than the same truck equipment for 12-in. tires, there is a saving in weight of 1203 lb. per truck. The details of this reduction are given in Table XI.

TABLE XI

	Lb.
Weight of 5-ton truck axle, exclusive of brake-drum	1,660
Weight of two 1½-ton solid-tire axles, exclusive of brake-drum	1,200
Saving in axle weight	460
Saving in weight of tires	520
Total saving	980
Less excess wheel and brake-drum weights	77
Net saving in weight due to use of 8-in. tires	903
Saving in weight of radius-rods, springs, etc.	300
Net total saving in weight	1,203

As compared with a solid-tired truck of equal weight capacity, we have the savings given in Table XII.

TABLE XII

	Lb.
Weight of two 48 by 12-in. solid tires and wheels	1,262
Weight of four 40 by 8-in. pneumatic tires and wheels	1,208
Excess weight of solid-tire equipment	54
Saving in weight of axle	460
Saving in weight of springs, radius-rods, etc.	300
Total saving over solid tires	814

The area of contact of four 8-in. pneumatic tires upon the road is about 27 per cent greater than two 12-in. pneumatics. This additional surface keeps the tires from sinking in soft places, gives better traction under conditions when most needed and in ordinary service the additional area gives better traction. As compared to solid tires in winter service, off of paved roads, etc., the four pneumatic tires have all of the advantage.

The tandem rear drive has about the same advantageous effects over single-axle construction that the pneumatic would have over solid tires. With the tandem combination, when passing over an obstruction in the road the chassis is raised only one-half the distance it would be in the regular type of construction. This reduces the acceleration of bodies upon the chassis to one-fourth that with ordinary construction. Thus, by reducing shocks and vibration, the number and cost of repairs due to fatigue of metal, etc., are reduced by a large percentage.

The tandem construction makes for such exceptional riding qualities that a glass filled with water to within 1 in. of the top, attached to the rear of the six-wheel truck, lost none of its contents even when operated over a decidedly rough and rocky road.

The most destructive factors of the operation of vehicles upon pavements are the wheel-load and the wheel-thrust. Therefore, if one of two tires under the rear end of a standard 5-ton truck bears down upon the road with a pressure of 8000 lb., it will find a great many places where the road will allow it to crush through and once the break is accomplished a rut or chuck hole is started. However, if this single tire is replaced by two that are separated a considerable distance, the pressure of each of the smaller tires becomes 4000 lb. upon the pavement instead of 8000 lb. and a place in the road that will break through is seldom found. Fig. 17 indicates how chuck holes get started by excessive wheel-loads, when applied to a weak spot in the pavement.

This tandem rear-drive cuts in two the heavy wheel-loads and the thrust as well, thereby making it possible to design pavements suitable for 3½-ton trucks, and carry 5 and 7-ton loads upon them with no increase in their destruction.

The tandem-axle combination has a decided advantage over both the regular pneumatic and solid-tired types in that four brakes of 21-in. diameter are available in place of two brakes of 21-in. diameter, the maximum possible with the two-wheel rear-drive type.

Pneumatic tires permit of an increase in the average speed that is just double that of solid tires and the tandem rear-drive will permit increased minimum speeds on bad roads, rough streets, etc. In view of the fact that the trucks now building are for the purpose of demonstrating these points, I am very desirous that you gentle-

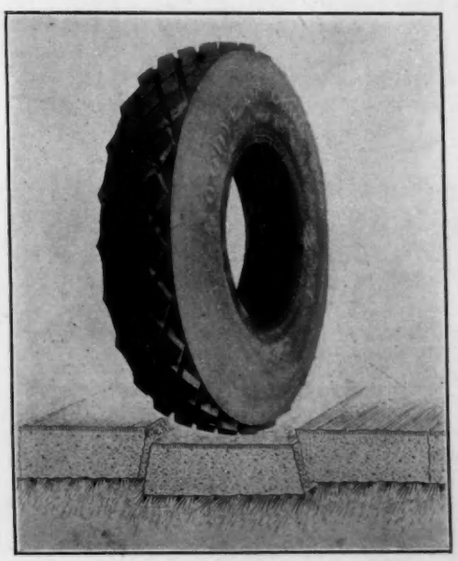


FIG. 17

men suggest tests that you consider would prove the points that are yet open for proof.

SUMMARY

The design of the ultimate motor truck to use pneumatic tires will in general require the following changes over present solid-tire design:

(1) Considerably higher rear-axle gear-reductions, ranging from 5 to 8 for pneumatics as against 7 to 16 for solid tires
(2) About 15 per cent increase in engine sizes
(3) Lower transmission low-gear reductions, ranging from 9 to 14 for pneumatic as against present solid-tire practice of 3 1/3 to 6. This will automatically introduce a fifth speed, which will account for better average road speed
(4) Lighter unsprung parts such as front and rear axles and wheels
(5) Multiple-wheel construction to accommodate the heavier tonnage, securing the advantages of pneumatic-tire equipment

FEBRUARY COUNCIL MEETING

THE following were in attendance at the meeting of the Council held at the offices of the Society on Feb. 13: President J. G. Vincent, First Vice-president J. G. Utz, Past-president Charles M. Manly and Councilors David Fergusson, F. M. Germane and N. B. Pope. Considerable discussion was had as to the time and place of holding the next Summer Meeting of the Society. Chairman Beecroft of the Meetings Committee reported that Ottawa Beach, Mich., where the 1919 Summer Meeting was held, was, according to a straw vote the committee had taken, most favored for the coming meeting, there being also considerable sentiment among the members that the meeting should be held at Saratoga Springs, N. Y. As to the week during which the meeting should take place, about the same time as last year seemed to be the most satisfactory to the members. Additional points in connection with the coming S. A. E. gathering referred to the making of exhibits, whether members only should be allowed to take part in the sports and each member should be restricted to a specified number of guests. The Meetings Committee was empowered to arrange for the exhibition at the meeting of anything new of technical value in the way of exhibits, not of an objectionable commercial nature, with a view to informing the members of engineering advances in which they are particularly interested.

The Council gave advice as to the securing of technical papers on specific subjects to be presented at the meeting. Definite announcement as to these and other features of the meeting will be made to the members as promptly as possible.

Ninety applications for membership were approved, eighty-eight of these being for individual membership and two for affiliate membership. Carl H. Amon was transferred from Junior to Member grade.

It was reported that seventy-one applications for membership were received during January of this year, as compared with seventy-four during January of last year; further, that the membership of the Society included on Jan. 31, 1920, 4386 individual and affiliate members and that there were in addition on that date 108 affiliate member representatives and sixty-three enrolled students.

The following subjects were assigned to the Divisions of the Standards Committee indicated:

Ball Studs—Miscellaneous Division.
Carriage Bolts—Miscellaneous Division.
Clutch Release Forks—Transmission Division.
Lamp Voltage for Electric Vehicles—Electric Transportation Division.

The Council discussed briefly the amendments to the Constitution proposed at the meeting of the Society held in January, it being understood that a further study of the points involved would be made before the March Council meeting with a view to some specific recommendations being made then. The Council favored generally the proposed amendments as to the vacating of elective offices of the Society in the case of officers being unable to attend to their prescribed duties.

A Tractor Engine Test

By C. A. NORMAN[1] (*Member*) AND B. STOCKFLETH[2] (*Non-Member*)

CHICAGO TRUCK AND TRACTOR MEETING PAPER *Illustrated with* PHOTOGRAPH AND CHARTS

THE tests published here were carried out with an engine of lower speed and of lower compression ratio than those used by Prof. O. C. Berry in the tests reported before this Society and elsewhere. More factors are covered; notably jacket loss, exhaust temperature, cooling through the carbureter, vacuum in the inlet manifold, etc. Attempts were made to take indicator diagrams and, at lower speeds, some interesting information was obtained. As the indicator used proved unreliable at higher speeds, this phase of the experiments will not be referred to here.

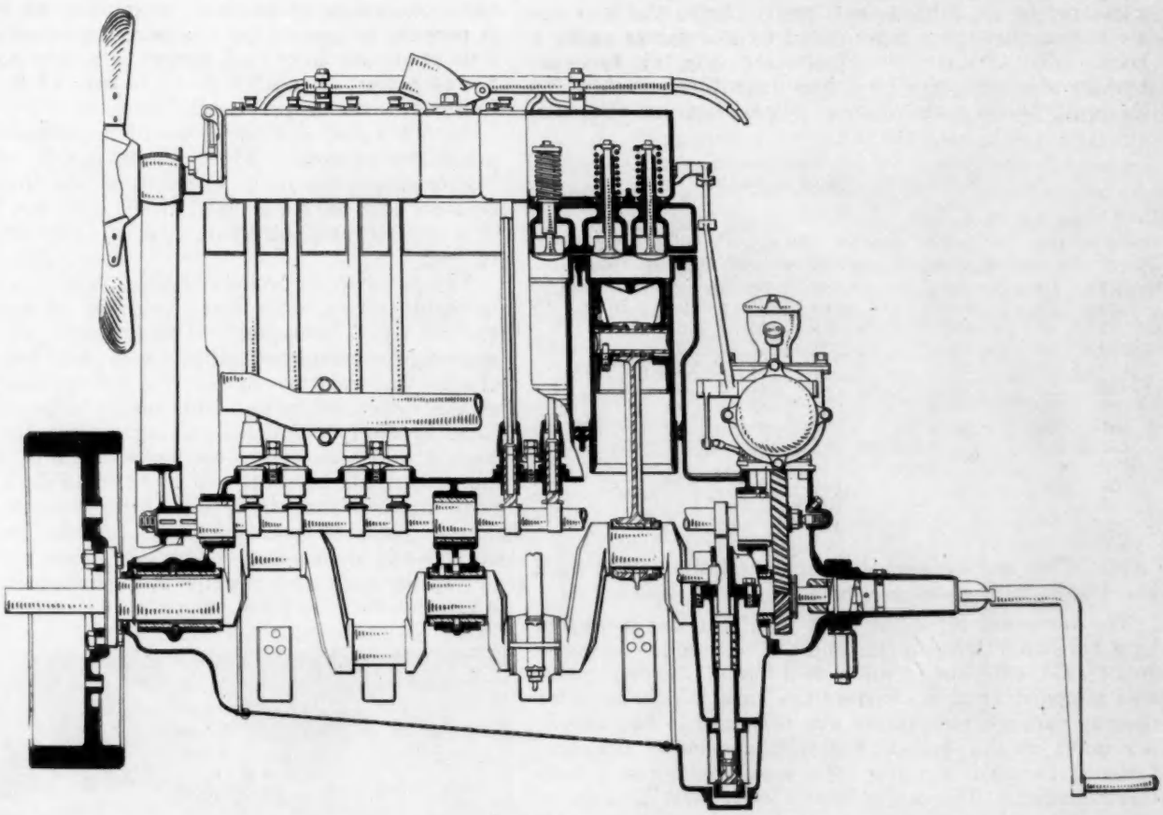


FIG. 1

The engine used, which is shown in Fig. 1, was a four-cylinder 4 by 5-in. truck and tractor engine intended to run on kerosene as well as on gasoline, and therefore had the very low volumetric compression ratio of 3.36. It had been in use in the field and at the factory for experimental purposes for over 2 years and, apart from a thorough cleaning and valve grinding, received no special treatment for these tests. It was found possible only by suitable adjustments to make it show a fuel consumption as low as 0.67 lb. per b. hp.-hr. However, with hardly any variation in power and only by a different carbureter adjustment, the fuel consumption at 600 r.p.m. increased to approximately 1.2 lb., an increase of 70 per cent. This would seem to demonstrate strikingly the importance of knowing what constitutes the best adjustment for an engine and of disseminating such knowledge among the engine using public.

The inlet valve had a 0.225-in. lift, while the lift of the exhaust valve was 0.285 in. Both valves had the same opening, 1⅝ in. On an average the inlet valve opens at 12 deg. after the dead center and closes at 40 deg. after the dead center. The exhaust valve opens 49 deg. before the dead center and closes 11 deg. after the dead center. The fuel used in the series of tests was Red Crown gasoline with a specific gravity of 0.733, a sample fractional distillation of which is on the next page.

The higher heating value of this fuel was 19,288 B.t.u. per lb., giving a lower heating value of about 18,000 B.t.u. per lb.

The general arrangement of experimental apparatus is shown in Fig. 2. The air in passing to the engine first enters an equalizing tank of about 17-cu. ft. capacity, through a 2-in. sharp-edged orifice specially calibrated against a standard gas meter from the Peoples Gas,

[1] Professor of machine design, Ohio State University, Columbus, Ohio.

[2] Chief engineer of experimental laboratory No. 5, International Harvester Co., Chicago, Ill.

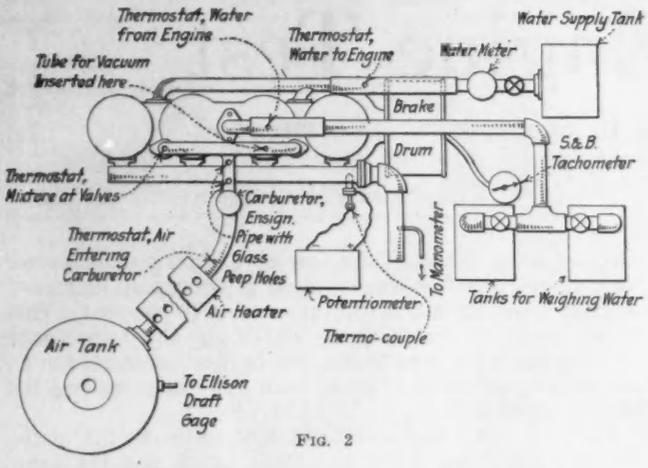


FIG. 2

Light & Coke Co. The vacuum in the tank, which is equal to the pressure drop through the orifice, was measured by an Ellison draft gage. From the tank the air passes through a pipe heated by gas flames under a hood. The temperature of the air entering the carbureter was measured by a bare bulb thermometer. An Ensign type JK carbureter, of 1¼-in. nominal size, was used. This was connected to the intake manifold by a short pipe, with glass peep-holes.

100 CC. SAMPLE USED IN DISTILLATION

cc.	Temperature, deg. fahr.	Check Temperature, deg. fahr.	Westphal Specific Gravity	cc.	Temperature, deg. fahr.	Check Temperature, deg. fahr.	Westphal Specific Gravity
5	156	152	.....	55	276	271	....
10	173	169	.....	60	290	284	0.746
15	187	180	0.679	65	305	294	....
20	194	194	.....	70	317	308	....
25	204	205	.....	75	332	320	0.766
30	214	216	0.703	80	348	339	....
35	230	228	.....	85	367	354	....
40	239	239	.....	90	399	387	0.789
45	253	249	0.728	92	408	.....	....
50	266	262		Loss—6.5 cc. Residue—1.5 cc.			

The temperature in the inlet manifold was measured by a bare bulb thermometer and the vacuum in the manifold by a U-tube, using water as a liquid. Cooling water was supplied from a constant-pressure tank, and after passing through the jackets was delivered to two weighing tanks on the scales. The inlet and outlet temperatures of the cooling water were measured by bare bulb thermometers. The temperatures were held practically constant by controlling the water flow by a valve. All the thermometers were calibrated against a thermometer certified by the Bureau of Standards. The temperature of the exhaust was taken by a Leeds & Northrup recording instrument, using a base-metal thermocouple.

The indicator drive was taken off the magneto shaft as best illustrated on the photograph shown in Fig. 3. The power was absorbed by a prony brake.

METHOD OF TESTING

The engine was run as far as possible at a series of set speeds with the throttle locked in a wide-open position, except at 300 r.p.m. where slight throttling was necessary. At each engine speed, 300, 400, 500, etc., r.p.m., the temperature of the air to the carbureter was held successively at certain fixed temperatures, usually 70, 85, 100, 120, 160, 200 and 240 deg. fahr.

After the establishment of the desired temperature, the carbureter was set first for the maximum richness of mixture at which the engine would run satisfactorily, then for the minimum richness. Thereafter, five or more intermediate settings were used. Every carbureter setting was held for 30 min. and readings taken every 5 min. At least 49 half-hour runs were thus made at each speed. The engine was cleaned and the valves touched up after the completion of the series of runs at each speed to maintain it in an unchanged condition and thus make certain of truly comparable results.

In Figs. 4 to 11 inclusive are given eight sets of curves showing for each speed the influence of the richness of the mixture and of the air temperature on the power developed. The same influence on fuel economy is shown for the two speeds in Figs 12 and 13. At lower speeds the power, as will be seen, is influenced very little by variation of richness, between rather wide limits. Up to 400 r.p.m., practically no variation appears throughout the whole range of workable mixtures. At 500 r.p.m. it is possible to operate the engine on mixtures of less than 8 lb. of air per lb. of fuel, though at a very marked sacrifice of power. From 9.5 lb. up to over 17 lb. the power, however, remains practically constant.

At 600 r.p.m. a drop in power is noticeable with air quantities exceeding 14 to 15 lb. per lb. of fuel. At higher speeds the range of constant and optimum power becomes ever narrower until at 900 and 1000 r.p.m. there is a decided peak at about 10.5 to 11 lb. of air per lb. of fuel.

This behavior is probably best explained by the greater ignitability and more rapid progress of combustion of the over-rich mixtures. Theoretically, 1 lb. of fuel requires for complete combustion slightly less than 15 lb. of air. This mixture, however, is not so readily ignitable, as the richer mixtures. This seems to be further indicated by the fact that the amount of heat dissipated into cooling water, as well as the temperature of the exhaust, increases quite regularly up to 13 to 15 lb. of air. The greater heat dissipation is probably due to longer-sustained high temperatures, as combustion continues through the whole expansion stroke, giving a greater heat flow to the walls and perhaps even partial after-burning in the exhaust. If these values were plotted the drop of

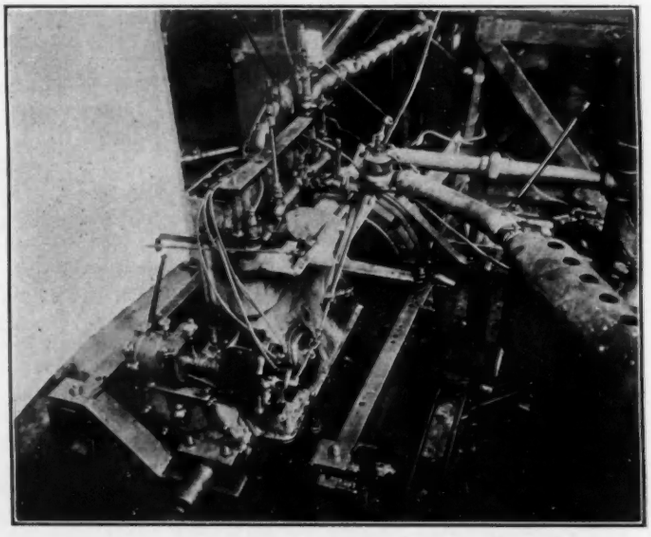


FIG. 3

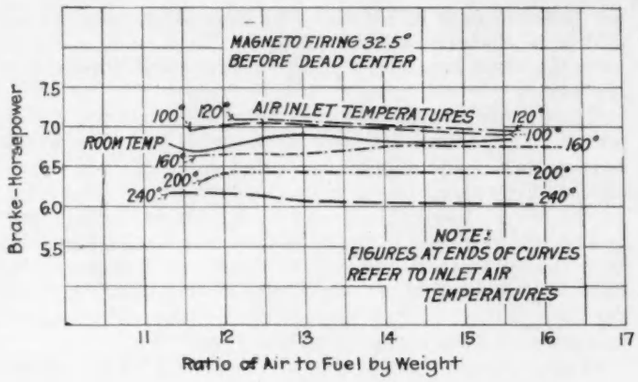

MAGNETO FIRING 32.5° BEFORE DEAD CENTER
AIR INLET TEMPERATURES
ROOM TEMP
NOTE: FIGURES AT ENDS OF CURVES REFER TO INLET AIR TEMPERATURES
Brake-Horsepower
Ratio of Air to Fuel by Weight

FIG. 4
Speed 300 r.p.m.

the curves, which is especially apparent in the curve for the exhaust temperature at air quantities over 15 lb., is due no doubt to lower temperatures generally, on account of air excess and less heat produced by combustion.

The influence of increased richness of mixture on fuel

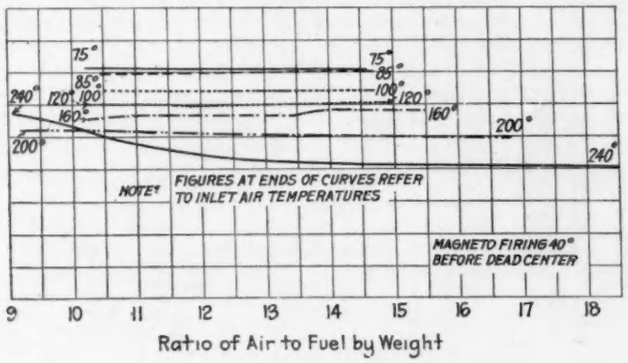

NOTE: FIGURES AT ENDS OF CURVES REFER TO INLET AIR TEMPERATURES
MAGNETO FIRING 40° BEFORE DEAD CENTER
Ratio of Air to Fuel by Weight

FIG. 5
Speed 400 r.p.m.

economy is uniformly toward a greater fuel consumption. This is only natural, as part of the fuel must go through unburnt or burnt to carbon monoxide. (See Table I, which gives some very rough determinations in this respect.)

At 900 and 1000 r.p.m., very lean mixtures likewise show an increase in the fuel consumption per brake horsepower due to a low power development with the combustion too slow for the speeds. This increase, however, is slight and should not be dwelt upon to the engine using public. Over-richness is the great source of fuel waste and should be preached against.

The general effect of air heating as shown by the

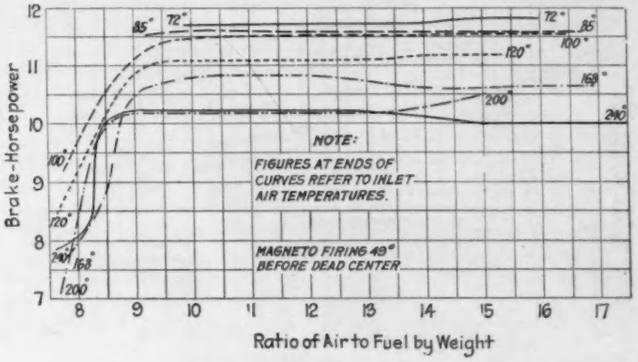

NOTE: FIGURES AT ENDS OF CURVES REFER TO INLET AIR TEMPERATURES.
MAGNETO FIRING 49° BEFORE DEAD CENTER
Brake-Horsepower
Ratio of Air to Fuel by Weight

FIG. 6
Speed 500 r.p.m.

curves in Figs. 4 to 11 is to reduce power. The amount of this reduction from the maximum to the minimum performance at some selected air to fuel ratios is shown in Table II and varies from 5.48 per cent, at 800 r.p.m. and the 15 to 1 ratio, to 15.7 per cent, at 400 r.p.m. and the 15 to 1 ratio, but can on an average be said to vary between 10 and 15 per cent.

This reduction in power can be satisfactorily explained by the increase in compression work and the decrease in the maximum pressure due to the increased temperature. The decrease is not wholly an unmixed evil, however. The engine can, at an increased air temperature, be made to operate between wider limits of fuel to air ratio or, to show greater flexibility and a smoother performance. Moreover, the fuel economy, especially at the higher speeds, is not affected very much by variations in the air temperature. As a matter of fact it has been found, as

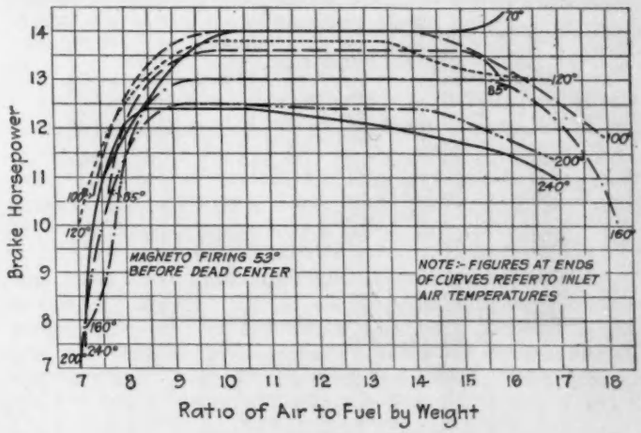

MAGNETO FIRING 53° BEFORE DEAD CENTER
NOTE:- FIGURES AT ENDS OF CURVES REFER TO INLET AIR TEMPERATURES
Brake Horsepower
Ratio of Air to Fuel by Weight

FIG. 7
Speed 600 r.p.m.

TABLE I

Speed, r.p.m.	Air Inlet Temperature, 70 deg. fahr. — Mixture Ratio of Air to Fuel by Weight	Percentage of CO_2	Percentage of CO	Mixture Ratio of Air to Fuel by Weight	Percentage of CO_2	Percentage of CO	Air Inlet Temperature, 240 deg. fahr. — Mixture Ratio of Air to Fuel by Weight	Percentage of CO_2	Percentage of CO	Mixture Ratio of Air to Fuel by Weight	Percentage of CO_2	Percentage of CO
300	.....	....	....	.....	....	....	15.85	8.5	4.5	11.31	4.2	9.5
400	15.09	11.0	1.2	10.68	5.8	7.7	18.99	6.8	0.3	9.53	4.3	10.6
500	15.96	9.4	0.3	9.85	5.0	8.0	17.44	4.8	0.4	7.58	0.3	[1]1.0
600	15.89	7.3	0.1	7.50	3.0	8.0	16.88	9.7	0.2	7.07	2.0	9.6
700	15.98	2.0	[1]0.3	7.65	2.3	8.4	17.79	6.3	0.4	7.37	1.7	8.0
800	13.90	7.7	3.4	6.83	3.2	11.9	18.39	5.3	1.0	6.86	3.8	11.2
900	13.78	9.5	2.8	7.65	4.2	11.5	18.59	8.4	0.3	6.78	2.6	10.4
1,000	15.60	7.3	1.4	7.02	3.8	12.1	19.54	6.4	0.5	6.54	2.4	9.5

[1]Sample diluted with air, due to leakage of stop-cock on sample can.

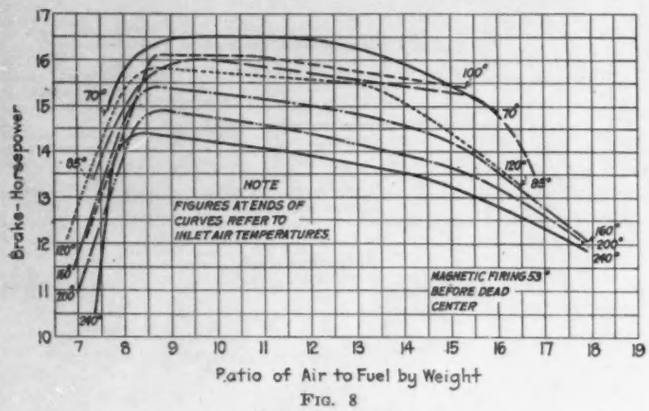

Brake-Horsepower
Ratio of Air to Fuel by Weight
NOTE
FIGURES AT ENDS OF CURVES REFER TO INLET AIR TEMPERATURES
MAGNETIC FIRING 53° BEFORE DEAD CENTER

FIG. 8
Speed 700 r.p.m.

stated by H. L. Horning at the 1919 Semi-Annual Meeting of the Society, that air heating will in actual road work give improved fuel economy even though the reverse may have been true on the engine test stand. This is due, no doubt, to the more positive carburetion and therefore better distribution with heated air.

GENERAL PERFORMANCE CURVES

In Fig. 15 is given a set of general performance curves of the engine as derived from these tests. They refer to air temperatures of 72 and 240 deg. fahr. respectively and to a mixture ratio of 12 to 1. This ratio seems to give the most consistent power development throughout the range.

The engine was an old-type engine designed for use in an "8-16" tractor. It will be noticed that the brake horsepower of the engine at a rated speed of 1000 r.p.m. exceeded the rated pulley horsepower by more than 5 hp. at the lowest temperature and by 2.5 hp. at the highest.

The fuel economy at a ratio of 12 to 1 varies but little with the speed from 400 to 1000 r.p.m. It is to be noted, however, that a fuel economy fully 20 to 25 per cent better than that at 12 to 1 can be obtained by using weaker mixtures as shown in the full-line curve.

In view of the importance of knowing just where the best performance occurs, there are given in Table III, the air temperature and mixture ratio leading to the greatest economy at each speed. It is of interest to note that here the highest air temperatures are very much in evidence. In other words, while rich mixtures and low temperatures

TABLE II

Speed, r.p.m.	Power, b.hp.	Temperature of Air Entering Carbureter, deg. fahr.	Ratio, Air to Fuel, by Weight	Decrease in b.hp. from about 70 to 240 deg. fahr., per cent
300	6.80	75	15 to 1	11.00
	6.05	240		
400	9.55	75	15 to 1	15.72
	8.05	240		
500	11.76	72	15 to 1	14.95
	10.00	240		
600	13.80	70	15 to 1	14.82
	11.75	240		
700	15.45	70	15 to 1	14.25
	13.25	240		
800	15.50	85	15 to 1	5.48
	14.65	240		
900	19.30	120	12.5 to 1	9.34
	17.50	240		
1,000	21.80	70	11 to 1	14.70
	18.60	240		

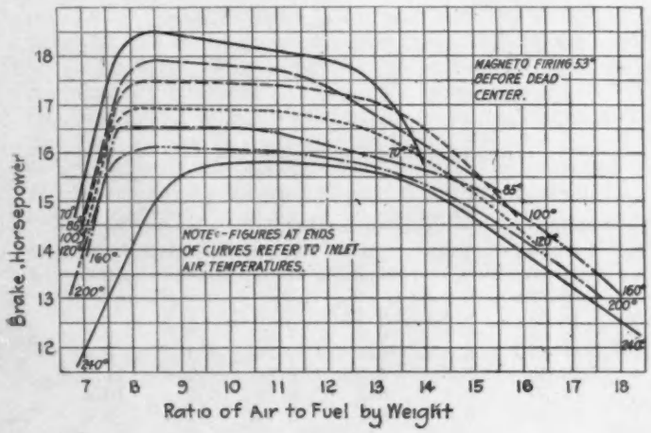

Brake Horsepower
Ratio of Air to Fuel by Weight
MAGNETO FIRING 53° BEFORE DEAD CENTER.
NOTE:-FIGURES AT ENDS OF CURVES REFER TO INLET AIR TEMPERATURES.

FIG. 9
Speed 800 r.p.m.

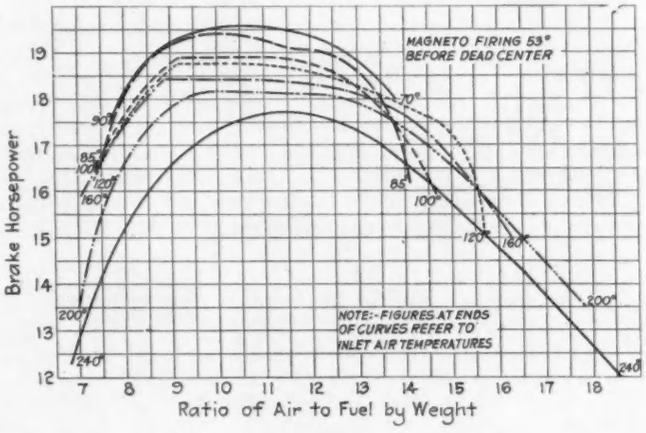

Brake Horsepower
Ratio of Air to Fuel by Weight
MAGNETO FIRING 53° BEFORE DEAD CENTER
NOTE:-FIGURES AT ENDS OF CURVES REFER TO INLET AIR TEMPERATURES

FIG. 10
Speed 900 r.p.m.

tend to give maximum power, leaner mixtures and higher temperatures are conducive to the greatest fuel economy. It is a question whether this fact, in connection with the smoother operation at higher temperatures, should not be the guiding factor in engine design rather than simply a desire for maximum power. Another point of interest is that the greatest economy occurs at speeds of from 400 to 600 r.p.m., rather than at higher speeds.

In Fig. 16 is shown a set of curves indicating that the percentage of cooling loss is very little less at 400 to 600 r.p.m. than at the higher speeds. On the other hand, the curve in Fig. 17 shows that the exhaust temperature increases with the speed. The indications are that the combustion, even with comparatively rich mixtures, is too slow to be fully utilized for power purposes at the higher speeds.

It is interesting to note that any increase in the air temperature does not influence either cooling loss or exhaust temperature at all. (See Figs. 18 and 19.) This leads to the conclusion that the mixture just prior to combustion, although at a slightly different temperature, is almost in the same state of evaporation in all cases. Since, as shown later, we have positive evidence that an air temperature of 240 deg. fahr. gives practically complete evaporation of the fuel before it reaches the cylinders, we can assume that even at low air temperatures the fuel is perfectly evaporated before combustion com-

TABLE III

Speed, r.p.m.	Fuel Consumption, lb. per b.hp.-hr.	Temperature of Air for Greatest Economy, deg. fahr.	Ratio by Weight, Air to Fuel, for Greatest Economy
300	0.85	160	16.25 to 1
400	0.68	240	19.00 to 1
500	0.67	240	17.80 to 1
600	0.68	160	18.25 to 1
		120	16.60 to 1
700	0.72	100	15.30 to 1
	0.74	70	16.00 to 1
		160	
		200	17.75 to 1
		240	
800	0.75	240	18.30 to 1
			18.00 to 1
900	0.78	240	15.70 to 1
1,000	0.80	240	16.00 to 1

mences; it is evaporated during the intake and the compression strokes.

Interpretation of Results

It is interesting to inquire how far the results obtained could have been quantitatively predicted on the basis of known physical laws; or to what extent can an engine of unusual size or new type be designed and its performance predicted on the basis of advance calculations?

In some fields of prime movers, especially that of steam turbines, it is customary to predict the performance by an intelligent analysis of the physical occurrences known to

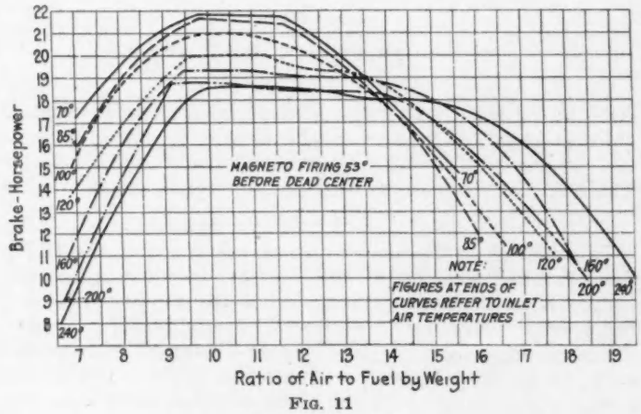


Fig. 11
Speed 1000 r.p.m.

take place from step to step and, by making proper allowances, estimate the future performance to within very close limits. It has been found possible to do very much the same thing in connection with continuous-combustion gas turbines. In this way a great deal of unnecessary, and often hopeless, experimentation has been avoided. Can we then follow a similar course in the case of an automotive type internal-combustion engine? There are many reasons, especially the uncertain character of the combustion process with over-rich mixtures and the complex character of some of the losses, why a very high degree of exactness cannot be expected. Nevertheless, it may be useful to see just which factors appear to remain unsolved after a reasonable attempt has been made to find the answer.

Assume an air temperature of 85 deg. fahr. and a ratio of air to fuel of 15 to 1 or practically the theoretical ratio for complete combustion without excess air. The fuel will have a lower heating value of 18,000 B.t.u. per lb. The air passing through the carbureter and the inlet

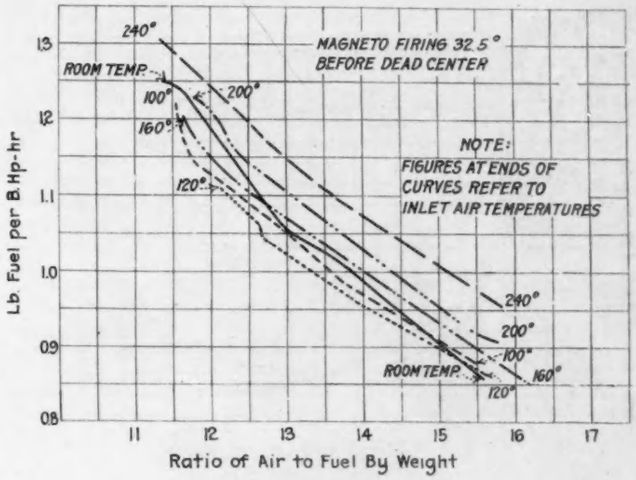


Fig. 12
Speed 300 r.p.m.

manifold to the cylinder is subject to a cooling effect due to the evaporation of the fuel and to a heating effect caused by radiation from the exhaust manifold and the cylinder walls. The cooling effect can be calculated in advance on the basis of the specific heat of the liquid, usually about 0.4 to 0.5, and the heat of evaporation of the same.

The total heat absorbed from a fuel temperature of say 70 deg. fahr. to complete evaporation may be put roughly at 230 B.t.u. per lb. of fuel, or in our case $230/15 = 15.3$ B.t.u. per lb. of air. With a specific heat of 0.24 for air, the cooling effect would be

$$\frac{15.3}{0.24} = 64 \text{ deg. fahr.}$$

As shown in Fig. 20 this is the actual cooling effect observed at a speed of 400 r.p.m. for a mixture of 15 lb. of air per lb. of fuel at an air temperature of 240 deg. fahr. At higher speeds the heating effect from the exhaust manifold is greater and the cooling through evaporation may be somewhat less. However, the curves in Fig. 20 show also that with an air temperature of 85 deg. fahr. there is no appreciable cooling of the mixture in passing through the carbureter. This means that the evaporation at this temperature is very incomplete, as can, for that matter, be clearly perceived by looking at the mixture through the peep-holes close to the car-

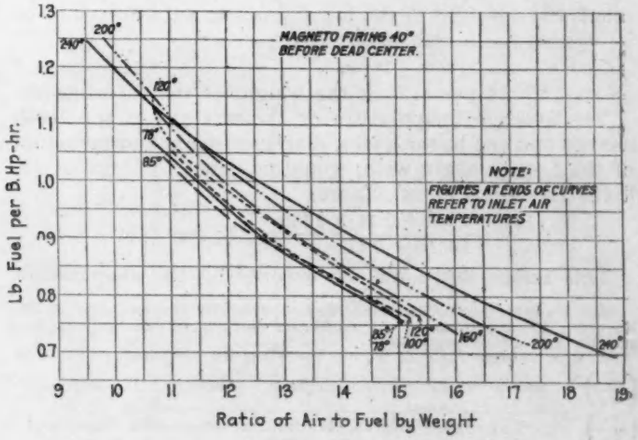


Fig. 13
Speed 400 r.p.m.

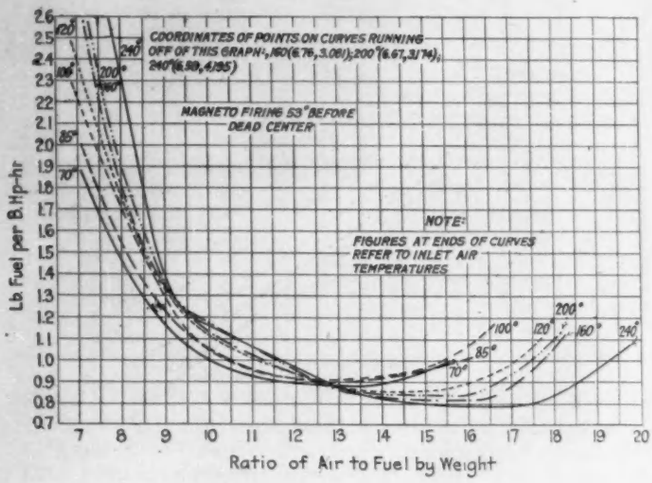

COORDINATES OF POINTS ON CURVES RUNNING OFF OF THIS GRAPH: 160(6.76, 3.08); 200°(6.67, 3.74); 240°(6.59, 4.35)
MAGNETO FIRING 53° BEFORE DEAD CENTER
NOTE: FIGURES AT ENDS OF CURVES REFER TO INLET AIR TEMPERATURES
Lb. Fuel per B.Hp-hr
Ratio of Air to Fuel by Weight

FIG. 14
Speed 1000 r.p.m.

bureter; also that the effect of heating the manifold may be very appreciable.

Avoiding all speculation as to any heating in the inlet manifold and in the inlet valve, assume that the mixture enters the cylinder at 85 deg. fahr. It here comes in contact with the burnt gases remaining in the clearance space at the end of the exhaust stroke. The temperature of these gases, as will be seen later, may be assumed to be around 1600 deg. fahr. The volume of the clearance space is 42.3 per cent of the piston displacement. The weight of the burnt gas may be assumed to be

$$\frac{(460+85)}{(460+1600)} \times 0.423 = 11.2$$

or roughly 11 per cent of the weight of the entering air.

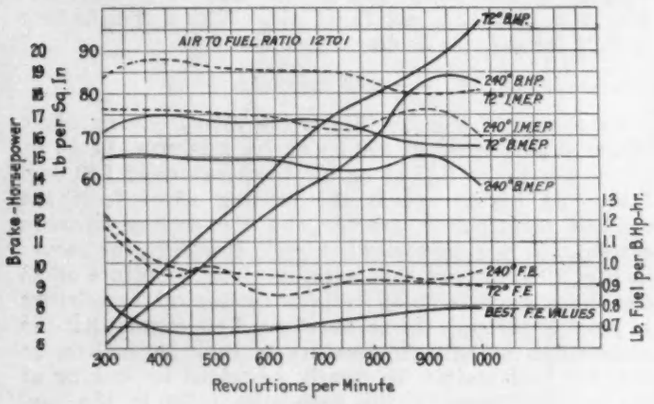


FIG. 15

The average temperature of the mixture of the fresh charge and the burnt gases, disregarding the evaporation of the fuel to begin with, would be, since for 1 lb. of air there are 16/15 lb. of charge,

$$\frac{16/15 \times 85 + 0.11 \times 1600}{16/15 + 0.11} = 227 \text{ deg. fahr.}$$

This temperature will be lowered by the evaporation of the fuel in the hot cylinder but will be increased by the radiation from the cylinder walls, which may have a temperature at least in the clearance space of 700 deg. fahr., or more.[4] What the resultant effect will be can be estimated only on the basis of observations. From indicator diagrams taken at 400 r.p.m. it appears that the heating by radiation from the walls preponderates.

As a starting point for further calculations, the temperature at the beginning of compression has been taken as 275 deg. fahr. With the customary compression exponent of 1.3 and a volumetric compression ratio of 3.36, the end temperature of compression will be

$$1.44\ (460+275) = 1060 \text{ deg. fahr. absolute or } 600 \text{ deg. fahr.}^5$$

With a lower heating value of 18,000 B.t.u. per lb. of fuel, there are available per pound of mixture

$$\frac{18000}{16} = 1125 \text{ B.t.u.}$$

The combustion is, however, by no means instantaneous.

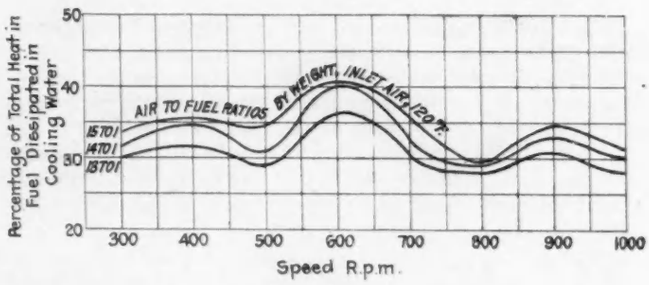

AIR TO FUEL RATIOS BY WEIGHT, INLET AIR 120°F.
15 TO 1
14 TO 1
13 TO 1
Percentage of Total Heat in Fuel Dissipated in Cooling Water
Speed R.p.m.

FIG. 16

Assume 70 per cent of the heat to be developed practically at constant volume in the clearance space, 30 per cent to go to after-burning and to cover cooling losses. The heat then remaining in the clearance space is

$$0.7 \times 1125 = 787 \text{ B.t.u. per lb. of mixture.}$$

In a paper[6] by the senior author presented at the 1919 Semi-Annual Meeting of the Society, is given a set of curves showing the variation with the temperature of the specific heat of the products of combustion. From the curve for specific heat at constant volume the average value to be used here is estimated at 0.236. The temperature increase in the clearance space will thus be

$$\frac{0.787}{0.236} = 3340 \text{ deg. fahr.}$$

The temperature reached will be

$$600 + 3340 = 3940 \text{ deg. fahr.}$$

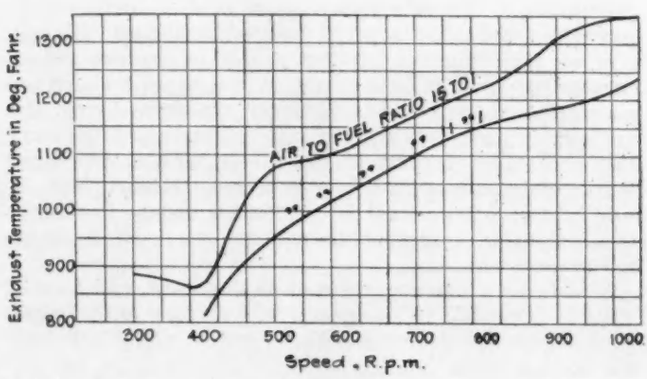


FIG. 17

[4] See Dugald Clerk, The Gas, Petrol and Oil Engine, and C. A. Norman, *Transactions* of the American Society of Mechanical Engineers for 1918, page 745.

[5] With a volume ratio of V_2/V_1, the absolute temperature ratio is $\left(\frac{V_1}{V_2}\right)^{n-1}$ where n is the compression exponent. In our case $\left(\frac{V_1}{V_2}\right)^{n-1} = 1.44$. (See Marks, Mechanical Engineers' Handbook, page 318)

[6] See THE JOURNAL, July, 1919.

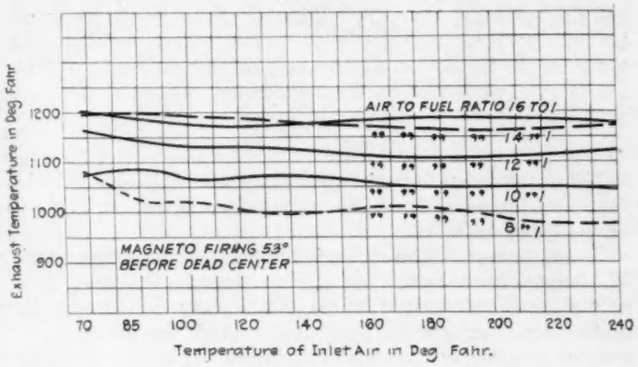

Exhaust Temperature in Deg. Fahr
AIR TO FUEL RATIO 16 TO 1
14 " 1
12 " 1
10 " 1
8 " 1
MAGNETO FIRING 53° BEFORE DEAD CENTER
Temperature of Inlet Air in Deg. Fahr.

FIG. 18

The nature of the expansion from the maximum pressure is, of course, uncertain. It will be found, however, that the cooling loss to be expected is of the order of magnitude of the heat assumed to be developed by after-burning. In other words, the gases would expand adiabatically.

Under such conditions the expansion exponent may be taken from the author's paper previously mentioned from the curves for C_p/C_v and will for the temperature range expected be about 1.255. The temperature ratio is then 1.36.

The end temperature of expansion is

$$\frac{3940+460}{1.36} - 460 = 3240 - 460 = 2780 \text{ deg. fahr.}$$

We might now check the assumed cooling loss by a

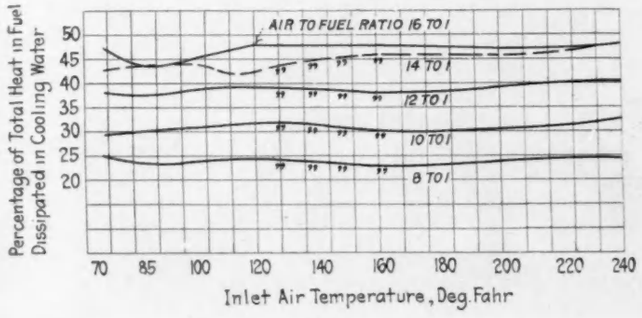

Percentage of Total Heat in Fuel Dissipated in Cooling Water
AIR TO FUEL RATIO 16 TO 1
14 TO 1
12 TO 1
10 TO 1
8 TO 1
Inlet Air Temperature, Deg. Fahr

FIG. 19

formula published by the senior author in a paper[7] previously referred to.

The formula is

$$L = 3.2\,\frac{CR}{N}\sqrt{V} \text{ where}$$

L = cooling loss during the expansion stroke in B.t.u. per lb. of gas

R = the ratio of cooling surface to enclosed volume, sq. in. to cu. in., in our case 1.29

V = the piston speed in feet per minute; in our case, at 400 r.p.m., $V = \frac{2 \times 400 \times 5}{12} = 335$ ft. per min.

C = a constant obtained by the author from certain experiments by Dugald Clerk and is read from curves in Fig. 21, which is reproduced from the original paper. The temperature referred to is the mean temperature during the expansion stroke

[7]See *Transactions* of the American Society of Mechanical Engineers for 1918, page 745.

It is in our case

$$\frac{3940+2780}{2} = 3360 \text{ deg. fahr.}$$

With exterpolation for this temperature, C may be taken to be 1600 B.t.u. per lb. of gas.

The cooling loss is consequently

$$L = \frac{3.2 \times 1600 \times 1.29\sqrt{335}}{400}$$

$= 302$ B.t.u. or $\frac{302}{1125} = 26.8$ per cent of the fuel heat.

This value is at least of the same order of magnitude as the 30 per cent reserved for after-burning in our assumptions.

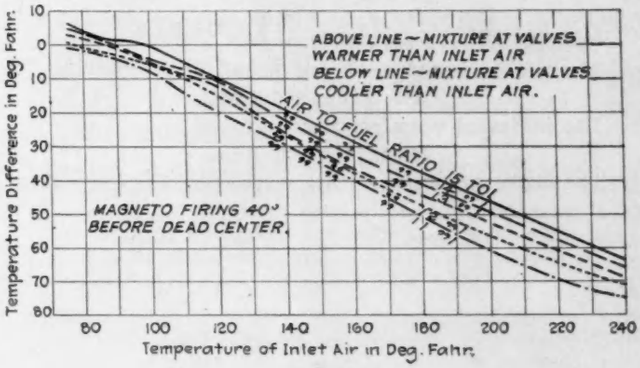

Temperature Difference in Deg. Fahr.
ABOVE LINE ~ MIXTURE AT VALVES WARMER THAN INLET AIR
BELOW LINE ~ MIXTURE AT VALVES COOLER THAN INLET AIR.
AIR TO FUEL RATIO 16 TO 1
MAGNETO FIRING 40° BEFORE DEAD CENTER.
Temperature of Inlet Air in Deg. Fahr.

FIG. 20

The net indicated work obtained from the process is the difference between the expansion work and the compression work per pound of gas.

The formula for both is

$$\text{Work per pound in B.t.u.} = \frac{RT_1\left(1-\frac{T_2}{T_1}\right)}{778 \times (n-1)}, \text{ where}$$

R is a gas constant equaling 53.3 for air and 53.6 for average combustion gas

T_1 is the absolute initial and

T_2 is the absolute final temperature

n is the exponent of change, equaling 1.3 for compression and 1.255 for expansion in this case

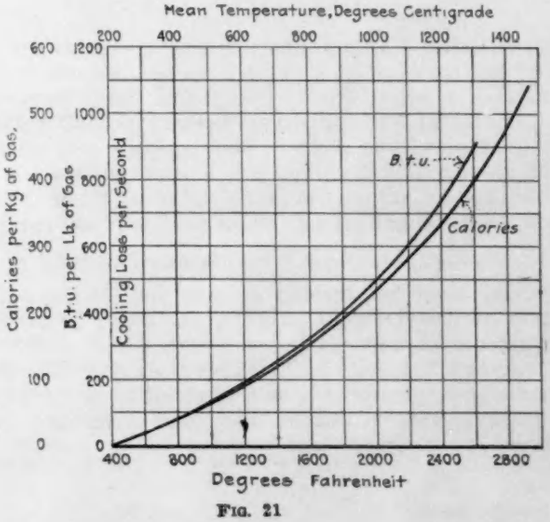


FIG. 21

Speed, r.p m.	Fuel Consumption, lb. per b.hp -hr.
300	0.86 (throttling)
400	0.76
500	0.71
600	0.76
700	0.76
800	0.80
900	0.86 (exterpolated)
1,000	0.96

The compression work per pound of air, neglecting the very small compression work on the fuel vapor, is

$$\frac{53.3 \times 735}{778 \times 0.3}(1 - 1.44) = 74 \text{ B.t.u. or}$$

$$74 \times 15/16 = 70 \text{ B.t.u. per lb. of mixture}$$

The expansion work per pound of mixture is

$$\frac{53.6 \times 4400}{778 \times 0.255}\left(1 - \frac{1.00}{1.36}\right) = 315 \text{ B.t.u.}$$

The net indicated work per pound of mixture is

$$315 - 70 = 245 \text{ B.t.u.}$$

The indicated work per pound of fuel is

$$245 \times 16 = 3920 \text{ B.t.u.}$$

The brake net work, with a mechanical efficiency actually determined for this engine at about 0.85, is equal to $3920 \times 0.85 = 3330$ B.t.u. One horsepower hour is equal to 2547 B.t.u. Consequently the amount of fuel necessary per brake horsepower-hour is

$$\frac{2547}{3330} = 0.765 \text{ lb.}$$

The actually measured values for the conditions of richness and temperature here assumed are given in the accompanying table.

It will be noticed that from 400 to 700 r.p.m. the check is surprisingly good. At higher speeds the fuel rate is higher than figured here. This may be caused by throttling through the carbureter and the valves; by greater heating of the charge through radiation from the exhaust manifold and from the cylinder walls; by insufficient velocity of combustion, and by other factors, all of which the authors hope to be able further to analyze later on, with the aid of the indicator and entropy diagrams.

Work of this kind was started in connection with this paper and gave some very interesting indications. Unfortunately, the operation of the indicator used was not satisfactory enough to warrant publication of the results.

Internal-Combustion Engine Fuels[1]

By E. W. Dean[2] (*Member*) *Illustrated with* Photographs and Charts

ONE of the important factors in our present system of transportation is the use of vehicles propelled by internal-combustion engines. The use of automobiles and motor trucks has developed to such a stage that they can be regarded as practically indispensible, and people in general are looking forward to a wider use of this type of mobile equipment rather than to anything in the line of restriction. The fuel upon which these engines are dependent is, however, one that is obtained from a raw material which is drawn from comparatively limited natural resources.

The term "engine fuel" is a broad one, designed to include a wide range of products capable of furnishing power in internal-combustion engines. At present, however, "engine fuel" actually means gasoline, a product principally derived from crude petroleum. The production of crude oil has increased steadily throughout the period of years during which the automotive industry has been developed, but the rate of increase has been relatively small. The production of gasoline has increased much more rapidly than the production of crude oil, but even this has not been comparable with the growth of the use of motor-propelled vehicles. Consequently, a real problem exists in the matter of conservation of petroleum-derived engine fuels. The Bureau of Mines is endeavoring to promote measures favorable to more efficient production and is also working indirectly toward the development of satisfactory substitutes to be used in place of the present types of gasoline. This paper does not attempt to furnish an adequate discussion of the fuel problem nor is it devoted particularly to desirable measures in the direction of conservation. Its object is to provide information of general interest concerning the present engine-fuel situation with the hope that sufficient interest will be aroused to aid in supporting measures necessary for the more efficient utilization of one of the nation's most important resources.[3]

Before discussing the production of gasoline, it seems necessary to attempt some sort of a definition. This is not easy and the statement, given below, which is taken from the 1916 Webster dictionary, is hardly adequate for present purposes.

> Gasoline; a volatile, inflammable liquid used as a solvent for oils, fats, etc., as a carburetant and to produce heat and motive power.

I venture the following definition, as covering the present understanding of the word:

> Gasoline is an inflammable, relatively volatile liquid, derived from crude petroleum or similar in properties to petroleum derivatives, and containing a sufficient proportion of low-boiling constituents to permit the starting of an internal-combustion engine under moderately unfavorable conditions.

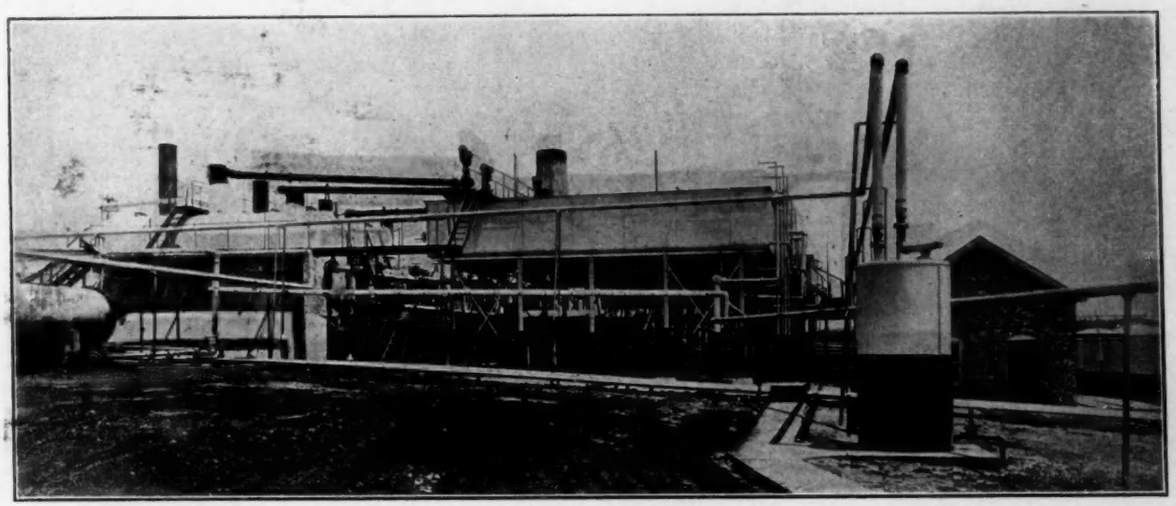


Fig. 1

The principal reason for offering this definition is to permit the making of a distinction between the words "gasoline" and "naphtha". Naphtha is a petroleum product that is generally, although not invariably, less volatile than gasoline, but which is chiefly distinguished by a lower content of low-boiling fractions. In other words, naphtha is gasoline minus the major portion of fractions distilling below the boiling point of water.

"Straight-Refinery" Gasoline Process

The oldest and even at present the most important method of producing gasoline is one involving distillation from crude petroleum. This method is frequently called the "straight-refinery" process and gasoline produced by it is termed "straight-refinery" or "straight-run" gasoline. The basic process is that of fractional distillation which involves the separation of products

[1] Paper, substantially in full, presented at a meeting of the Franklin Institute of Pennsylvania, Dec. 11, 1919.

[2] Petroleum chemist, Bureau of Mines Experiment Station, Pittsburgh, Pa.

[3] The material included in this paper has in large part been taken from the manuscript of a bulletin by H. H. Hill and E. W. Dean, which is to be printed at an early date.

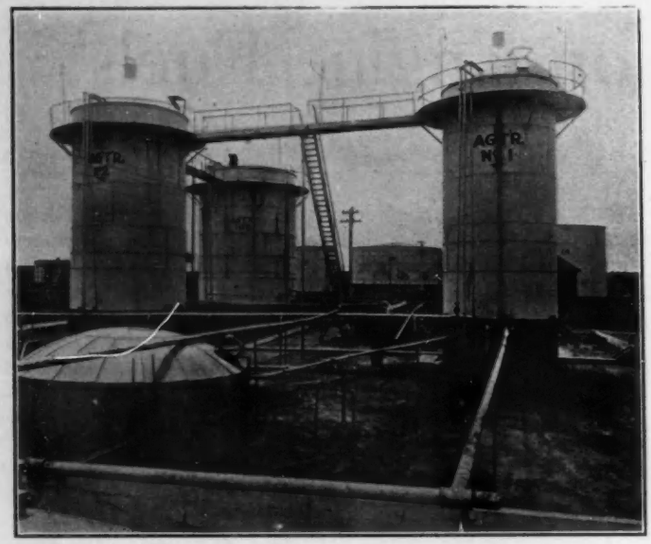


FIG. 2

having different ranges of boiling points. The process, as conducted in commercial practice, varies in a great many details and a discussion of these variations is beyond the scope of this paper. The following brief description covers perhaps the most common general procedure. Crude oil is distilled in cylindrical, horizontal stills, which are heated either by coal or oil. The vapors pass through condensers and the liquid products are run into receiving tanks. Fig. 1 shows a battery of crude stills with condensers and receiving tanks. The type of still represented in this illustration lacks the fractionating column employed in some of the more recent installations.

The product separated from the crude oil by this primary distillation is generally called "crude naphtha" or "crude benzine" in the refinery. The next process of treatment is usually a chemical one, principally intended to improve the color and odor of the product. Crude naphtha is pumped into lead-lined agitators and is there treated, first with sulphuric acid, then with a caustic-soda solution and finally with water. The stirring is usually accomplished by blowing air in at the bottom of the agitator tank. Fig. 2 shows three agitator tanks of this type. Another type less commonly used involves the so-called continuous process, in which crude naphtha passes through tanks containing respectively acid, water, alkali and water. The continuous method has some advantages, but according to my observation is less extensively employed than the non-continuous type involving stirring by air.

The final process in the refining of gasoline is a redistillation of the chemically-treated crude naphtha. This distillation is generally carried out in so-called steam stills. Steam stills are sometimes heated entirely by closed steam coils, and sometimes partly heated by fire while live steam is blown through the hot liquid. Steam stills are generally, although not invariably, equipped with fractionating towers and are often operated in batteries which permit continuous rather than "batch" distillation. Fig. 3 shows, in the foreground, steam stills of the type in which heat is supplied by closed steam coils. It will be observed that these stills are equipped with large fractionating columns.

The properties of the gasoline produced by the straight-refinery process are subject to considerable variation and are controlled in part by the physical and chemical nature of the crude oil from which it is derived and in part by the procedure employed in the refinery. Straight-refinery gasoline is composed chiefly of so-called saturated hydrocarbons, that is, hydrocarbons which are not

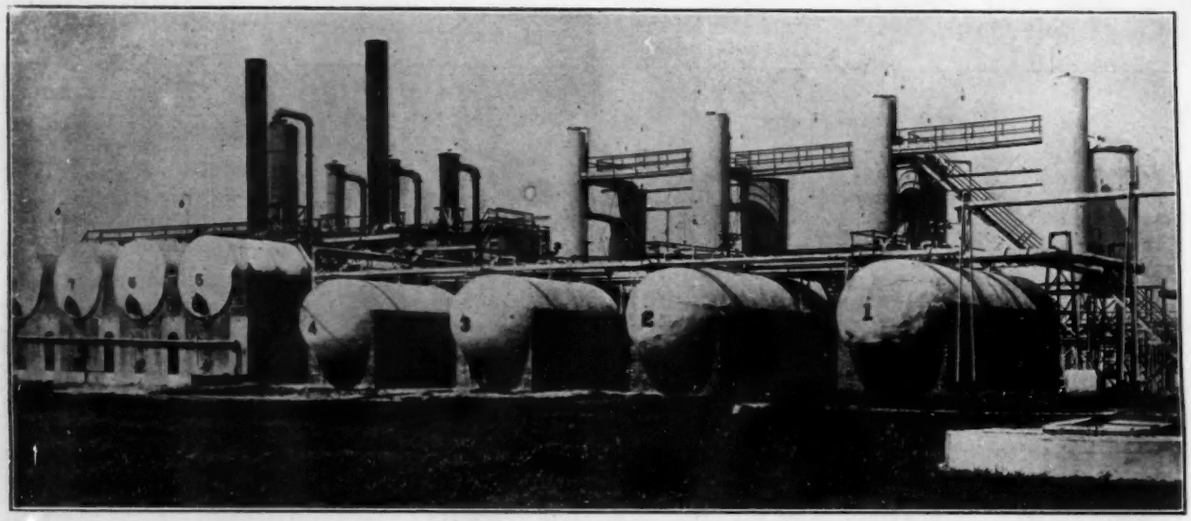


FIG. 3

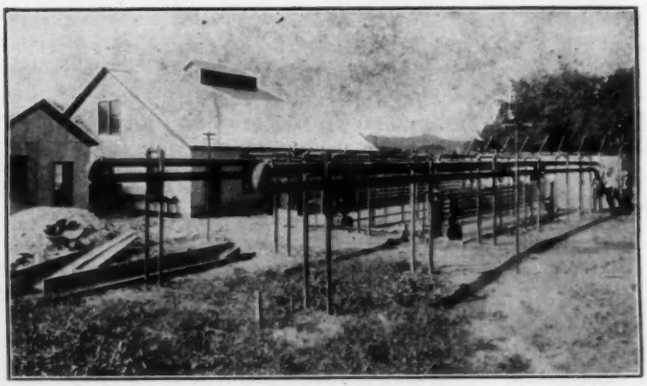


FIG. 4

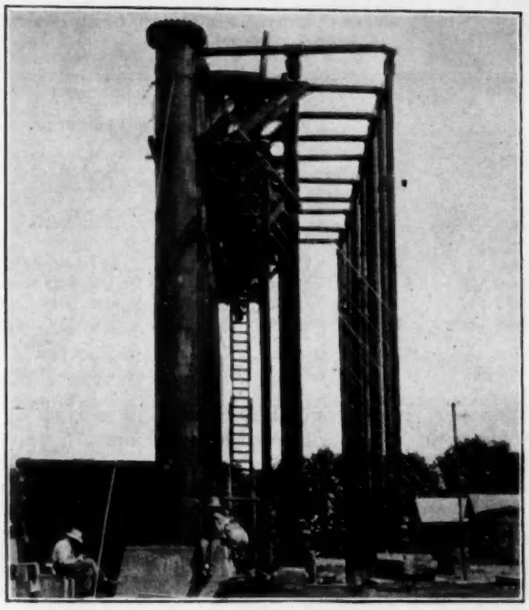


FIG. 5

chemically active. Most of the chemically active products which may be present in the crude naphtha from the primary distillation are removed by the chemical treatment with sulphuric acid. The property chiefly controlled by the refining process is the end point or upper boiling limit of the gasoline, which is fixed so as to separate the more volatile fractions from those generally marketed as kerosene or lamp oil.

"CASING-HEAD" GASOLINE PROCESS

A large proportion of the present supply of engine gasoline is derived not from crude petroleum but from natural gas. Gasoline from this source is commonly called "casing-head gasoline," due to the fact that before its commercial separation was in vogue it was discovered as a condensate around the heads of well casings. Natural gas frequently contains appreciable quantities of liquid hydrocarbons which, although of relatively low boiling point, can be readily separated. Two types of process to effect this separation are in general vogue, the older and perhaps still the more important being the so-called compression process. In this as usually operated the natural gas is compressed under a moderate pressure and then cooled. Subsequently it passes through a second set of pumps which brings it up to a high pressure. Cooling after this second compression effects the separation of the liquid hydrocarbons. This type of process is used only for natural gas of relatively high gasoline content. Fig. 4 represents a plant operating the compression process. Most of the equipment is enclosed and cannot be observed, but the cooling coils appear in the foreground.

The other process is the absorption method which involves washing the natural gas with either a heavy naphtha or a high-boiling petroleum oil, thereby dissolving out the bulk of the liquid hydrocarbons. When naphtha is used as the absorbent, a product is obtained from the absorption towers that is usually marketed without subsequent treatment. If a more efficient absorbent such as so-called straw oil or mineral seal oil is employed, the casing-head gasoline must be separated by distillation. Fig. 5 shows one of the first of the plants operating the absorption process.

Casing-head gasoline produced by the compression process is somewhat more volatile than that produced by the absorption method, but either product has too high a vapor pressure to permit safe handling or transportation.[4] The usual practice is to blend this gasoline either with naphtha or with high end-point gasoline, after which it can be shipped and handled with a reasonable degree of safety. Casing-head gasoline blends were unfavorably regarded by users when first put upon the market, but this was due more largely to improper methods of blending than to any inherent bad qualities of the product. The blended casing-head gasoline now marketed is probably rather more satisfactory in use than the so-called straight-refinery gasoline of equivalent average boiling point. This is on account of the greater ease with which a cold engine can be started with it. Statistics have shown that approximately 10 per cent of the nation's supply of gasoline is of the casing-head variety and, due to the necessity and desirability of blending with less volatile petroleum distillates, it is probable that the proportion of marketed gasoline containing products from natural gas is several times this figure.

CRACKING PROCESSES

Another recently developed method of producing gasoline is the so-called cracking process, which has received much publicity in the past few years. The basic principle of the process is one with which all chemists are familiar, namely, that if organic compounds are heated at sufficiently high temperature they decompose, with the formation of other compounds, a large percentage of which are of smaller molecular weight and lower boiling point. The cracking process has been used in the petroleum industry since the early days of its development, chiefly for the production of kerosene from the higher boiling fractions usually classed as fuel oil. This older process was conducted at atmospheric pressure. It appeared, however, that satisfactory yields of gasoline could not be obtained under these conditions as the hydrocarbons from which it is derived tended to vaporize before decomposing. To produce gasoline by the cracking process it is necessary to maintain the distillation system under pressure, thereby permitting the attainment of a high temperature before vaporization occurs. Cracking has also been effected by the scheme of passing liquid or vaporized oil through superheated tubes. Nu-

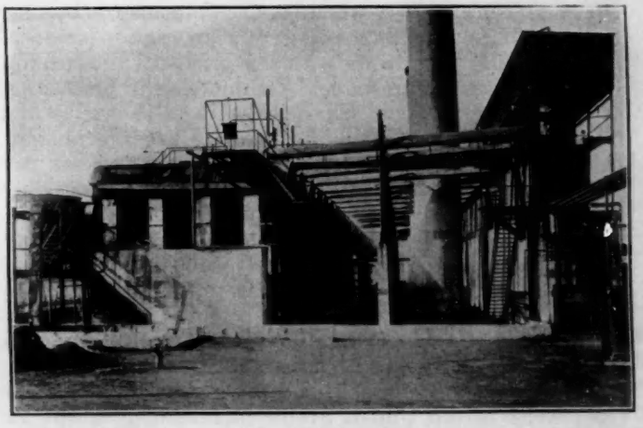


FIG. 6

[4]The Interstate Commerce Commission has issued regulations specifying the maximum allowable vapor pressure for gasoline.

numerous types of gasoline cracking processes have been developed, some of the pressure distillation type, others of the tube type. At present the pressure distillation type is the only one that has been developed on an important commercial scale. The process most commonly used is that of Burton, which was developed by the Standard Oil Co. of Indiana and which is operated by this company and by various others mostly of the Standard group.

Figures now available have indicated that approximately 10 per cent of the nation's supply of engine fuel is derived from cracking processes. Cracked gasoline is similar to straight-run gasoline as regards physical properties, but there is an essential chemical difference due to the fact that the cracking reaction produces a considerable percentage of olefin or unsaturated hydrocarbons. The content of these hydrocarbons is commonly estimated by observing the percentage of the gasoline absorbed by ordinary concentrated sulphuric acid. This does not accurately measure the content of olefins, but for general purposes, however, the term "percentage of unsaturation," meaning the proportion absorbed by sulphuric acid, is a convenient one to use. Untreated cracked distillates may have "percentages of unsaturation" as high as 40 per cent, a more usual figure, however, being 20 to 25 per cent. Cracked gasoline is usually marketed in the form of mixtures with straight-refinery and casing-head gasoline, these mixtures rarely being more than 12 per cent unsaturated, which indicates that the products of the cracking reaction are blended with at least an equal amount of casing-head or straight-refinery gasoline. Fig. 6 shows a battery of Burton stills. These stills are distinguished from the more common types of refinery equipment by the fact that they are operated under pressure and that they are equipped with special types of fractionating towers.

Gasoline Production Statistics

Statistical information is frequently uninteresting and, unless figures are carefully interpreted, it may be misleading. As regards the production and use of gasoline, however, the information now available is so definite and impressive that any misunderstanding is practically impossible. In the present connection it is interesting to present data regarding the magnitude of the nation's production and consumption of gasoline, the relation between production of gasoline and production of crude oil which is the principal raw material from which it is derived and the relation between production of gasoline and the magnitude of the most important outlet of consumption which is the internal-combustion engine. It is also interesting to note certain seasonal factors which have to do with the production, consumption and storage of gasoline. The increase in production of gasoline during the last few years is shown by the figures in Table 1. Assuming that the average refinery price of gasoline during the year 1918 was 18 cents per gal., it appears that the total value of the year's production of engine fuel is approximately $650,000,000.

An interesting relation can be observed by comparing the increase in the production of crude petroleum and gasoline and the increase in number of cars and trucks in use in the country. Fig. 7 represents these increases on a percentage basis, the 1909 figures being regarded in each case as the basic ones. It will be noted that, during the 10-year period, the production of crude oil from wells in the United States increased by about 95 per cent, the production of gasoline by approximately

Table 1.—Increases in the Annual Production of Gasoline in the United States

Year	Production of Gasoline	
	Bbl.	Gal.
1909	12,900,000	541,800,000
1916	49,020,967	2,053,880,614
1917	67,870,153	2,850,546,426
1918	85,007,451	3,570,312,942

560 per cent and the number of cars in service by over 1700 per cent. These figures bring out certain important facts. The production of gasoline has increased at a notably greater rate than has the production of crude oil. In 1909 the yield of gasoline from the crude refined was 10.7 per cent, whereas in 1919 the yield was 26.1 per cent. This increase in the relative amount of gasoline produced from the crude is accounted for in several ways. In the first place, the distillation range of gasoline has been notably changed, products now on the market being of considerably higher average boiling point than those in 1909. The production of casing-head gasoline and the use of cracking processes have been important factors in increasing the apparent yield of gasoline from crude oil. In addition, there undoubtedly have been increases in the efficiency of the conventional types of refinery equipment such as crude and steam stills. It may also be noted that the average quality of the crude oil produced in the country has probably improved as regards gasoline content. It is a question as to how near the present percentage of gasoline production per unit quantity of crude oil approaches the maximum that is commercially practical. It is impossible to make predictions regarding the future quality of crude oil. As regards the efficiency of refinery equipment, it has been estimated that this may be improved on the average by a maximum of not over 10 per cent. The production of casing-head gasoline undoubtedly has not reached a maximum, but here again it is difficult to attempt an estimate. The use of cracking processes certainly has not developed to the maximum possible extent and I have in the past ventured to predict that the quantity of gasoline produced from the crude could, if desired, be practically doubled if the maximum use were made of this means of production. These estimates have not been strenuously contradicted, but I am frank to admit that my own optimism has decreased considerably since they were offered. It is still believed that the use of cracking processes could double the production of gasoline, but it appears that this is not likely ever to be accomplished as there is a marked tendency for increases in the use of fuel oil, which is the product that must necessarily be sacrificed in case the maximum production of gasoline is attained through cracking. It is now believed that the chief limiting factor for the use of the cracking processes is likely to be the demand for fuel oil. As regards further increases in the end point of gasoline, considerable doubt exists. Many authorities in the automotive industry claim that the present limits are too high, although this probably means that they are too high for the types of engines that can and will be developed in the reasonably near future. The present heavy demand for kerosene indicates, moreover,

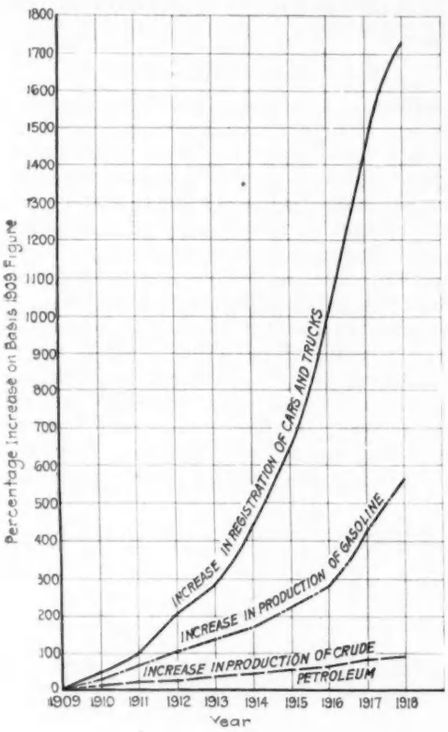

Percentage Increase on Basis 1909 Figure
INCREASE IN REGISTRATION OF CARS AND TRUCKS
INCREASE IN PRODUCTION OF GASOLINE
INCREASE IN PRODUCTION OF CRUDE PETROLEUM
Year

Fig. 7

that further changes in the quality of gasoline are not likely to occur in the immediate future as raising the end point of gasoline inevitably decreases the production of kerosene.

The fact that the number of cars in service has increased approximately three times as much as the production of gasoline is a fact meriting serious consideration. Although exact data on this point are not available, it is not believed that the average consumption of gasoline per car has notably decreased. The obvious inference is that in 1909 the percentage of gasoline used by automotive vehicles was considerably smaller than the percentage so used in 1918. Inasmuch as the use of gasoline and naphtha for other purposes than internal-combustion engines is increasing rather than decreasing, it is entirely reasonable to believe that at no very distant time the demand for gasoline is likely to exceed the supply, unless the producers of gasoline add to their already considerable accomplishments in production or the fuel efficiency of automotive equipment is greatly improved. It is realized that the production of gasoline in the calendar year 1919 has been slightly in excess of consumption, but this is readily accounted for by the fact that the petroleum industry increased its volume of production during the war, whereas the builders of automobiles and motor trucks have not yet gotten back to pre-war operating efficiency.

Aside from the statistical evidence indicating the necessity for conservation, it is interesting to observe certain facts regarding the seasonal variations in production, consumption and storage of gasoline. Fig. 8 is of interest, although it may be noted that the facts most clearly brought out are not those indicative of normal tendencies. As regards the production of gasoline it may be observed that there is a slight seasonal fluctuation, refineries working at greater efficiency in the summer than in the winter months. The consumption of gasoline varies notably, due to the fact that automobiles and motor trucks are less extensively used in cold than in warm weather. The quantities of gasoline stored naturally vary in the opposite direction, the stocks at the refineries increasing in the winter months and decreasing in the warm weather. The curves in Fig. 8 illustrate the general tendencies, although there are certain fluctuations in the maxima and minima which were caused by war conditions. It is of particular interest to note that stocks of gasoline reached an abnormally low mark in the late summer and early fall of 1918, there being at

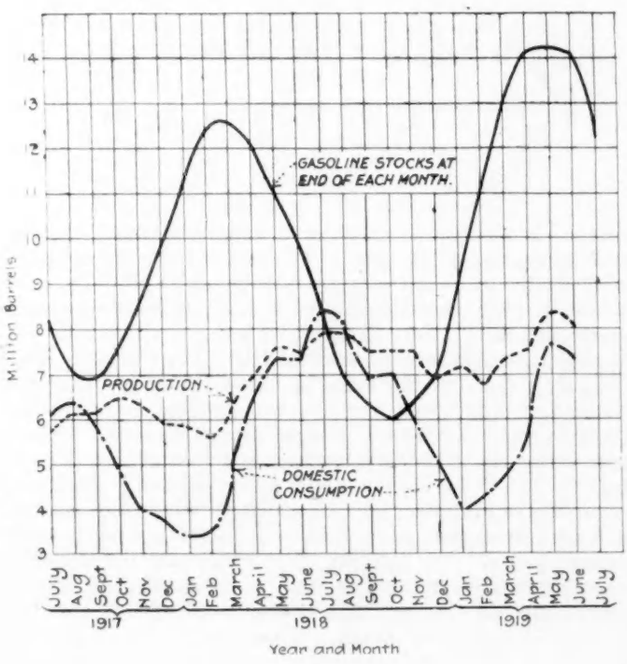


Fig. 8

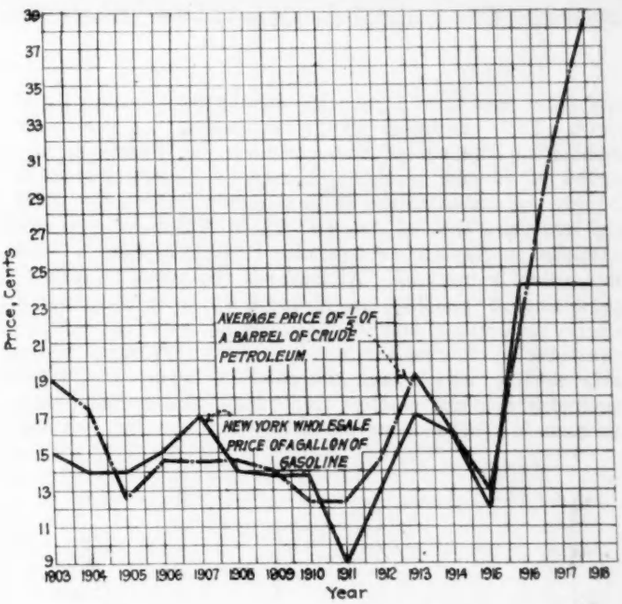


Fig. 9

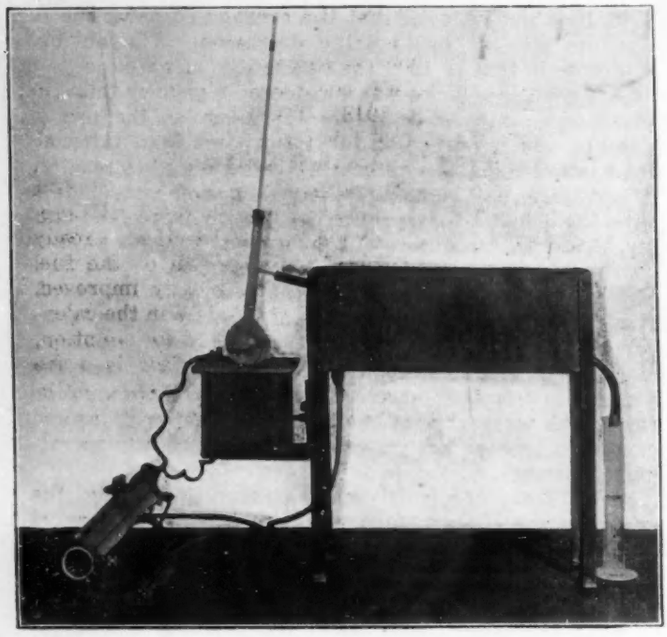


FIG. 10

this time less than 1 month's supply of gasoline in reserve. This condition was what caused the regulation prohibiting the use of motor vehicles on Sundays for a period of several weeks. It may be observed that although the consumption of gasoline in September, 1918, was less than that of August, which is normal, the consumption in September was also less than that in October, which is unusual. As regards the curve indicating quantities of gasoline in storage, it is interesting to observe an unusual situation for the present year. Maximum stocks of gasoline are usually in storage in the early spring, which was the case in 1918. In 1919 a maximum was not reached until May, and no important decrease in stocks occurred until July. In July of both 1917 and 1918, stocks were at a considerably lower level than in July, 1919.

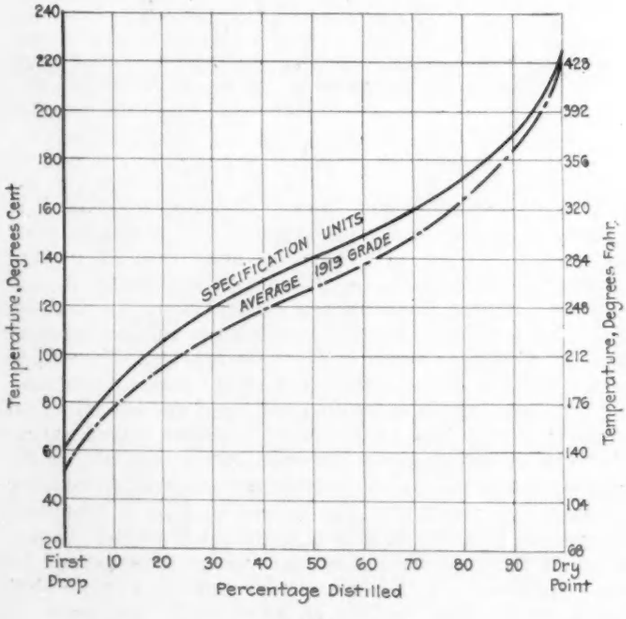


FIG. 11

MARKETING OF GASOLINE

The marketing of gasoline is a subject that is of particular interest, but does not, however, permit satisfactory discussion by a chemist. The following observations are presented with a certain degree of qualification. I feel that the basic data are reliable but do not guarantee my interpretation in all cases. Gasoline marketing practice obviously is not controlled by any simple physical and chemical laws, and any general tendencies that may be noted are principally emphasized by exceptions.

The marketing of gasoline generally involves jobbing and retailing organizations which handle the product after it is delivered by the producer. In some cases the three organizations, producing, jobbing and retailing, are entirely distinct, in others they are controlled and operated by a single company. Likewise there are cases where two of the organizations are under the same

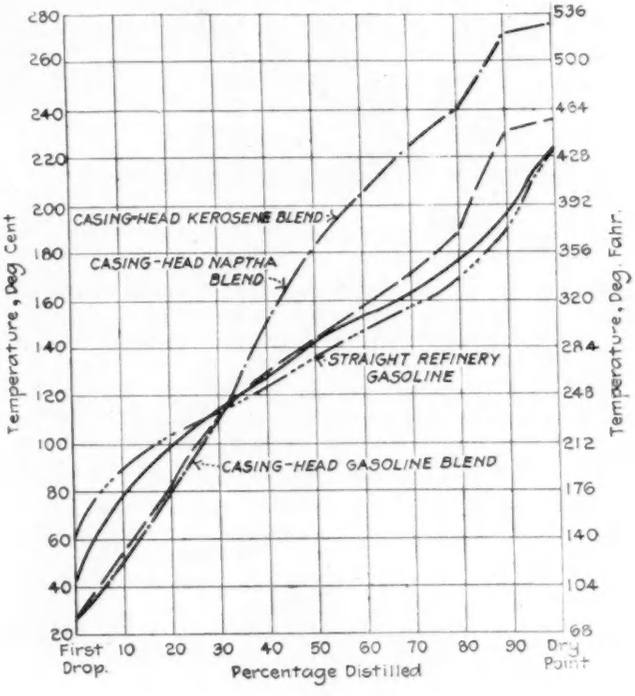


FIG. 12

general control and the other is separate. As a general rule competition in the retail end of the business is practically unrestricted, producing and jobbing companies selling gasoline freely to independent competing organizations. There are certain exceptions to this and it may be noted that in several cities in the Middle West the retail price is the same as the jobbers' price, which eliminates any profit for the independent retailer who tries to meet the general figure for the locality. This is not, however, a particularly serious matter, as the retailer's profit is always small and frequently does not cover losses and the cost of handling. The usual margin is 2 or 3 cents per gal., which is about 10 per cent of the selling price. In general, the small retailer handles gasoline chiefly for the purpose of attracting or retaining customers who purchase other commodities yielding a larger margin of profit. The larger filling stations probably make a fair profit on gasoline, although even

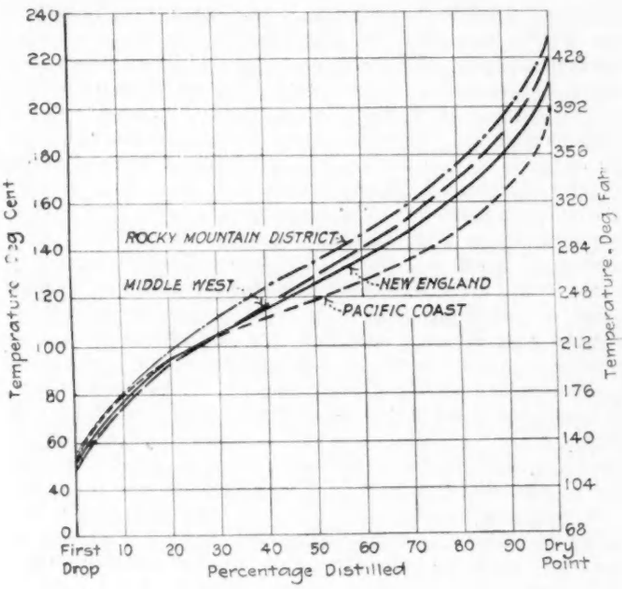


FIG. 13

here it is safe to assume that the really profitable branch of business is the sale of lubricating oil and occasionally other automobile accessories. It may also be mentioned that the system of retail distribution of gasoline is very highly organized and that customers receive service which is claimed to be the best maintained in any line of business. I am frank to admit that, with the possible exception of tobacco, I do not know of any commodity which is as generally available and efficiently dispensed as gasoline.

As regards the price of gasoline, there are no general rules which can be stated without bringing to mind a host of exceptions. The one generally recognized fact is that the jobbers' price for gasoline is generally that fixed by the larger marketing organizations. What actually determines the prices selected by these organizations is something which can be determined only in a very general way, and the chief point of importance to the general consumer is that competing organizations generally favor an upward rather than a downward revision.

The most important general factors regulating the price of gasoline are the costs of transportation from refining centers and the demands of customers as regards volatility range. Gasoline marketed at a considerable distance from centers of supply is more expensive than that marketed in the adjacent districts provided the grade demanded is the same. Likewise, districts that are satisfied with engine fuel of high end point generally get it at a more favorable price than similarly located districts favoring a more volatile type of gasoline. The figures in Table 2 show the retail prices maintained in the spring of 1919, in several cities throughout the country. It will be noted that prices were lowest in the Pacific States and in the Mid-Continent States nearest the producing centers and highest in the Atlantic Coast and Rocky Mountain States.

The most important thing to note in regard to the price of gasoline is its relation to the price of crude oil. Fig. 9 illustrates this in a decidedly satisfactory man-

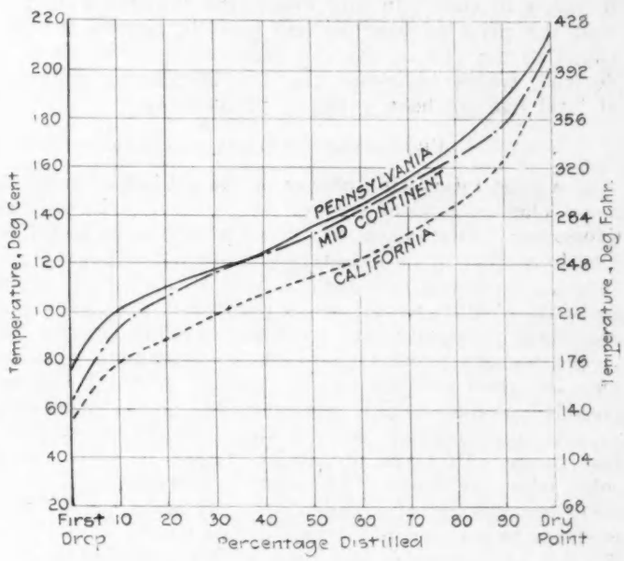


FIG. 15

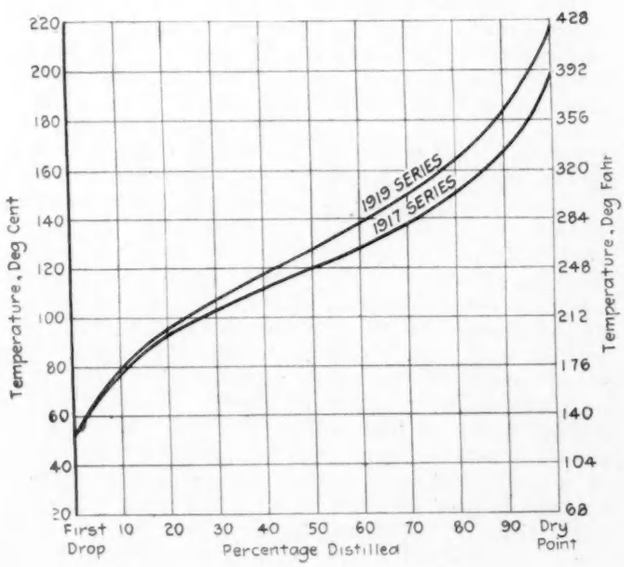


FIG. 14

ner. On this diagram are plotted New York jobbers' prices per gallon of engine gasoline and the average prices throughout the country of a fifth of a barrel of crude petroleum. It will be observed that the curves are in general parallel with the single notable exception for the year 1911, an explanation for which has not been worked out by the Bureau of Mines, although it seems possible that it may have been due to the fact that the Standard Oil Co. was ordered dissolved by Federal injunction at about this time. It may be noted also that since 1916 the price of gasoline has held practically constant, whereas the prices of crude petroleum has increased very considerably. This discrepancy is in part due to the fact that the New York price has held constant, whereas, in general, throughout the country, there has been an increase of 2 to 4 cents per gal. Even with this increase, however, there would be a tendency for the gasoline and crude oil curves to diverge. A probable explanation of this is the fact that an increased demand for other refinery products has permitted the charging of a lower percentage of operating costs against gasoline

TABLE 2.—RETAIL PRICES OF GASOLINE IN APRIL, 1919

State	City	Retail Price, Cents per Gal.
Massachusetts	Boston	27.5
New York	New York	27.0
Georgia	Atlanta	26.5
Illinois	Chicago	23.0
Kentucky	Louisville	25.5
Alabama	Birmingham	26.5
North Dakota	Fargo	27.1
Kansas	Wichita	22.7
Texas	San Antonio	24.5
Montana	Missoula	29.0
Colorado	Denver	28.0
New Mexico	Albuquerque	32.0
Washington	Seattle	21.5
California	San Francisco	20.5

than was customary in the earlier period. In the past year the supply of gasoline has slightly exceeded the demand which is another reason for the relatively slight increase in cost. In any event, the important fact is that the price of gasoline has been in general proportional to the cost of the raw material from which it is derived, which indicates that the oil-refining industry at least has not been guilty of profiteering.

PROPERTIES OF GASOLINE

The most interesting phases of the subject of gasoline for a chemist are, of course, its physical and chemical properties. Discussion of these, however, is likely to lead to a great many details which are not of general interest. Realizing this fact, I have attempted to avoid technical details, as far as is consistent with giving a reasonably adequate idea of what gasoline actually is.

The various properties which are commonly studied may be listed as color, odor, "doctor" test for decomposable sulphur bodies, corrosion and gum formation, gravity, unsaturation, acidity and volatility or distillation range. Of these properties it may be stated that color, odor, the "doctor" test and the corrosion and gumming test are not of sufficient interest and practical importance to warrant discussion in this paper. The gravity test is one which has been highly regarded in the past but which actually is of minor significance. This paper undertakes to show why it is of slight use in determining the quality of gasoline. The unsaturation test is of interest because it shows whether or not a gasoline contains products of the cracking reaction. The acidity test is one which covers care in refining and which has been included in Government specifications, although its practical value has not as yet been definitely determined. The most important property of gasoline is its volatility which is indicated by the determination of its distillation range. The behavior of gasoline in an internal-combustion engine is very largely determined by this property of volatility which is the basic one on which gasolines are graded by intelligent refiners. The older system of grading on the basis of specific gravity still holds to a limited extent, but even this is based upon the assumption that distillation range is a function of specific gravity, which assumption is correct only when considerably qualified.

The method for determining volatility involves an analytical distillation under regulated conditions. These conditions and the nature of the apparatus used have a considerable influence on the results and it is therefore necessary that tests be made in the same way, if figures are to be regarded as comparable. Testing methods were in the past subject to considerable discrepancy, but since the recent recommendation of a certain method by the Bureau there has been a marked improvement in this particular. Practically all specifications are now based upon a distillation by the method recommended by the Bureau of Mines, which has been described in various publications. The apparatus used in the test is represented in Fig. 10 and consists essentially of a source of heat, a 100-c.c. distillation flask of specified dimensions, an ice-cooled condenser of specified dimensions and a 100-c.c. graduate. The method involves heating the gasoline at a specified rate and reading the temperatures when the first drop falls from the end of the condenser, when each of the respective 10 per cent marks is reached and when the dry point is reached, the latter being taken as the highest temperature obtainable shortly after the flask has apparently boiled dry at the end of the distillation. Of these various readings, the following are usually sufficient to define the distillation range of gasoline: first drop temperature, 20, 50 and 90 per cent marks and the dry point. These are the points usually employed in specifications. Fig. 11 shows a curve plotted on the limits of the points set by the present Government specification for gasoline. It will be observed that each of the ends of the curve is rather definitely fixed by the use of two points. A point for the 50 per cent mark fixes the intermediate portion of the curve. This, however, is not absolutely necessary, as there are inconsiderable possibilities of variation in this point when the end limits are specified.

In discussing the properties of gasoline, a figure is sometimes employed which is called the average boiling point. This has not been employed in specifications as yet but is of some value in providing a single figure to be used as an index in stating the volatility of gasoline. This figure is obtained by adding the first drop temperature, each of the 10 per cent marks and the dry point, and then dividing by eleven. This figure provides the same sort of information that would be obtained by a specific gravity test, if the gravity test indicated what it is supposed by many to do.

DISTILLATION RANGE OF COMMERCIAL GASOLINE

Three methods of producing gasoline have been discussed, namely, the straight-refinery method, the casing-head gas method and the cracking method. Gasoline produced by the cracking process has no characteristic properties as regards distillation range. It is, however, frequently possible to distinguish gasoline blended with casing-head products from the straight-refinery types. Fig. 12 shows an interesting series of curves. One of the samples represented is a product made by blending casing-head gasoline with kerosene. Its distillation range is distinct from that of any of the other samples. This type of fuel is not regarded as entirely satisfactory. It may be noted, however, that neither the gravity test nor the average volatility figures would distinguish this gasoline from one of the more desirable types. The second sample of the series represented in Fig. 12 is one composed of casing-head gasoline, blended with a so-called heavy or high-boiling naphtha. This type of product is notably superior to the kerosene blend but is not entirely ideal as an engine fuel although gasoline of this type is used in large amounts and with reasonable satisfaction. It is believed, however, that the preferable type of casing-head blend is that represented by the third curve, which shows the distillation range of a sam-

ple made by blending a very moderate quantity of casing-head gasoline with the straight-refinery product. The curves for the three casing-head blends are shown in comparison with the curve for a straight-refinery gasoline. It will be observed that the casing-head blends have curves which for the first 20 or 30 per cent are lower than the straight-refinery product. Above this point, the curves are higher than the curve for the straight-refinery gasoline by varying amounts.

In 1917 and in 1919, the Bureau of Mines made surveys of the quality of gasoline sold throughout the United States. Certain results of these surveys are of general interest and are briefly discussed. In the first place, it was observed in the course of both surveys that the general quality of gasoline varied to a certain degree in various sections of the country. The Pacific Coast States received gasoline of relatively low boiling range, or high volatility. The Eastern States were next as regards low boiling range of gasoline. The Middle Western States received gasoline which was less volatile than that marketed on the Eastern coast, while the Rocky Mountain district received the least volatile grade of gasoline of any part of the country. Fig. 13 shows curves representing four different parts of the country, based upon the average of figures for the samples collected in each of the four districts. It will be noted that as regards the more volatile fractions of gasoline, represented on the lower end of the distillation curve, the differences are slight. Beyond the 20 per cent mark, however, the curves diverge and there are appreciable differences at both the 90 per cent mark and the dry point. A similar series of curves was obtained for samples collected in 1917, although at that time the Rocky Mountain district was not homogeneous as regards the quality of gasoline received. A certain proportion of gasoline marketed in this district came from California and was of the same general quality as that marketed on the Western coast.

The chief point of interest with regard to these curves is the general change in the quality of gasoline. The boiling range of gasoline has, in the past 5 or 10 years, been increasing notably. In the days when gasoline was a by-product of the refining industry, incident to the production of kerosene or lamp oil, its end point was in the general neighborhood of 300 deg. fahr. As the demand for gasoline increased, this end point has gradually raised. When the Bureau first began a study of the properties of gasoline, the figures usually maintained were between the limits of 350 and 400 deg. fahr. The end point has continued to increase, so that in 1919 the average was about 425 deg. fahr. As regards the change in quality between 1917 and 1919, the curves in Fig. 14 are instructive. These curves represent average figures for all of the samples collected in each of the respective surveys. It will be noted that the first drop and 20 per cent temperatures are practically the same in both cases but that the curves diverge beyond this point, the difference at the 90 per cent marks and dry points being between 15 and 16 deg. cent. This means that the gasoline marketed in 1919 was practically equal to that of 1917 as regards starting qualities but that the properties controlled by the end point had changed somewhat. Improvements in engine design have undoubtedly made up for a considerable proportion of this difference, and the gasoline marketed at present is probably used with fully as great satisfaction as that marketed in 1917. There is some question as to how much higher the end point can be raised and still yield satisfactory results. This is a point concerning which detailed information is lacking. The Bureau believes, however, that in general the grades of gasoline now on the market are being used with a reasonable degree of satisfaction. The practice of the refining companies is to avoid, as far as possible, dissatisfaction on the part of their customers, and if a certain grade of gasoline is subject to too many complaints it is practically certain to be modified. It is realized, however, that the demands of users in different parts of the country are not the same and that what is regarded as low-grade gasoline in one district would be looked upon as high-grade gasoline in another.

Gasoline Acidity and Unsaturation

In the test now recommended by the Bureau of Mines for determining the acidity of gasoline the residue left in the distilling flask after making a distillation is extracted with distilled water and tested with methyl orange indicator. Reddening of the indicator shows that the gasoline contains acid which was present in the original sample either free or in some sort of chemical combination. It has not been experimentally determined whether gasoline showing acidity according to this test has a particularly deleterious effect upon the metal parts of engines. It seems reasonable to suppose that it may have some undesirable effect, however, and inasmuch as this test is one which can be met by all refiners exercising reasonable care in preparing their products, it has been included in Government specifications. A study of the results of the Bureau's 1919 survey indicated that a majority of the samples were satisfactory according to this test.

The test for unsaturation is one which the Bureau has never recommended in specifications as there is reason to believe that the types of unsaturated gasolines usually marketed are equal, if not superior in some ways, to the saturated types. The unsaturation test is, however, interesting as showing the method by which gasoline was produced and is, therefore, worthy of brief discussion. The test employed by the Bureau simply involves agitating the gasoline with two volumes of ordinary concentrated sulphuric acid, and noting what per cent of the gasoline is dissolved. This test has been shown to cause a certain amount of polymerization of olefin or unsaturated hydrocarbons and on this account the percentage dissolved does not represent all of the olefins contained in the gasoline. As an empirical test, however, it has proved reasonably reliable and accordingly is of great value in the study of gasoline samples. The series of samples collected in 1917 varied in percentage of unsaturation from a fraction of 1 per cent up to a maximum of 6 per cent. For the 1919 series the lower limit was the same but a common maximum was in the neighborhood of 12 per cent, showing that companies operating cracking processes have practically doubled the proportion of cracked gasoline turned out.

As regards the properties of cracked gasoline, even when carefully refined, it has an odor that is more or less characteristic and which, to some at least, is slightly disagreeable. In addition, cracked gasoline is accused of the habit of polymerizing in shipment and storage with the deposition of gummy substances. Furthermore, cracked gasoline has been known to deposit gummy masses in the intake manifolds of engines and has also been accused of depositing carbon in cylinders. Some of these properties may be bad, although it is believed that in most cases the trouble is due to improperly re-

fined cracked gasoline rather than to the simple presence of cracked constituents. In any event, however, some of the Bureau's experiments have indicated a notable point of superiority. When ordinary gasoline is used in a high-compression engine, there is likely to be a "knock" which is undesirable. Cracked gasoline has been found to be less subject to this tendency than is the more highly regarded straight-refinery type. Frequent claims are made that cracked gasoline is superior to the straight-run type when cars are run up steep grades or worked under heavy load, and the Bureau's experiments with the phenomena of knocking have tended to substantiate these claims. In general, cracked gasoline is marketed in the form of blends which are not detected by the average user. It is believed, however, that even if the user were able to distinguish cracked gasoline, it would very likely be on account of its advantageous rather than its disadvantageous behavior.

Up to the last few years it was customary to rate gasoline on the basis of its Baumé gravity, and even today users are inclined to believe that a high Baumé gravity, which means a low specific gravity, is a desirable property. The specific gravity test, under certain conditions, does give an indication of the distillation range of gasoline, but in general furnishes highly unreliable information in this particular. It has already been pointed out that a mixture of kerosene and casinghead gasoline might have the same gravity as a straight-run gasoline. In addition it can be stated that the types of crude oil produced in different parts of the country have different physical and chemical properties and that for a given gravity there may be considerable difference in boiling range or vice versa. This fact is illustrated by the curves in Fig. 15. The three samples represented have practically the same gravity, but have distillation ranges which are appreciably different. The uselessness of gravity figures in indicating quality of gasoline is further indicated by the fact that engine fuels now on the market may contain benzol and other coal-tar derivatives which have abnormally low Baumé gravities.

Gasoline Substitutes

The subject of gasoline substitutes is one of particular interest, although definite information concerning it is decidedly limited. The first fact of importance to be recognized is that engine fuel is essentially a source of energy and that the energy must come from somewhere. Wonderful statements regarding the discovery of new fuels appear periodically in the newspapers, these fuels generally being made by adding a small quantity of some secret compound to water. That these discoveries are one and all without particular value is proved by the fact that water is a substance which contains about as little free energy as anything one could select.

As regards the real possibilities of gasoline substitutes, it now seems probable that oil shale may in the future prove an important source. There are enormous deposits of oil shales in the United States and the extraction of liquid oil from these is undoubtedly feasible, although it has not yet been worked out upon a commercial scale in this country. Oil shale will probably not assume any great importance as a source of engine fuel in the near future, but its potentiality as a resource gives a comfortable sense of assurance that the use of motor vehicles will not have to be discontinued when petroleum resources are exhausted.

Another type of fuel which offers unlimited possibilities for the future and which is already being developed to a certain degree is alcohol. The problems to be solved before this comes into general use are apparently the development of cheaper methods of production and the development of suitable types of engines. Efforts are at present being made to market a fuel containing alcohol and other components, which may be used satisfactorily in present types of internal-combustion engines. A sample of this fuel has been obtained by the Bureau in the retail market and has been subjected to laboratory examination. The distillation curve of this sample is shown in Fig. 16. Its exact chemical composition was not determined, but it was shown to contain both alcohol and benzol as well as a fair percentage of high boiling petroleum naphtha. The sample obtained by the Bureau was being sold at a price somewhat higher than that of engine gasoline. As regards its use, the Bureau has received reports from at least two reputable organizations indicating that it was found at least as satisfactory as ordinary gasoline.

The type of gasoline substitute which is of most importance at present is the mixture of hydrocarbons obtained as a by-product in the coking of coal. These so-called coal-tar distillates including benzol, toluol, xylol, etc., are hydrocarbons which are somewhat similar to the hydrocarbons found in petroleum, although of course there are well-recognized physical and chemical differences. During the war coal-tar distillates were largely used in the production of explosives and other munitions, but since the termination of hostilities a considerable supply has been rendered available for engine fuel. Several companies have already undertaken to market benzol mixtures and in the course of its 1919 survey the Bureau secured a number of samples which were sold as gasoline substitutes containing benzol. The distillation curve of one of these sample is shown in Fig. 16 and of some of the other samples in Fig. 17. The products are generally mixtures of coal-tar distillates with petroleum naphtha. These products have been used under widely varying conditions. In some cases favorable results have been obtained; in others, unfavorable results. Inquiries made by the Bureau have usually been answered by statements either that the substitutes were very satisfactory or very unsatisfactory, few of the users seeming to have found them about the same as petroleum gasoline. The use of coal-tar distillates as engine fuel does not, however, offer an ultimate solution of the engine-fuel problem as the total production of these hydrocarbons is rather limited. In discussing this subject in a paper prepared about 2 years ago, I had occasion to collect some statistics which are represented graphically in Fig. 18. This figure shows

(1) The United States' gasoline production in 1916
(2) An estimate of the total light-oil production, if all the bituminous coal mined in the country were coked in by-product ovens
(3) An estimate of the production of light oil, if all coking ovens were of the by-product type
(4) An estimate of the actual light-oil production for the year 1917

The actual figure for hydrocarbons suitable for engine fuel is even less than that for total light oil produced, and it appears, therefore, that as an engine-fuel resource coal-tar distillates are of relatively small importance. It can, for instance, be stated that since 1916 the production of gasoline has practically doubled, while the quantity of coal mined has increased only about 17 per

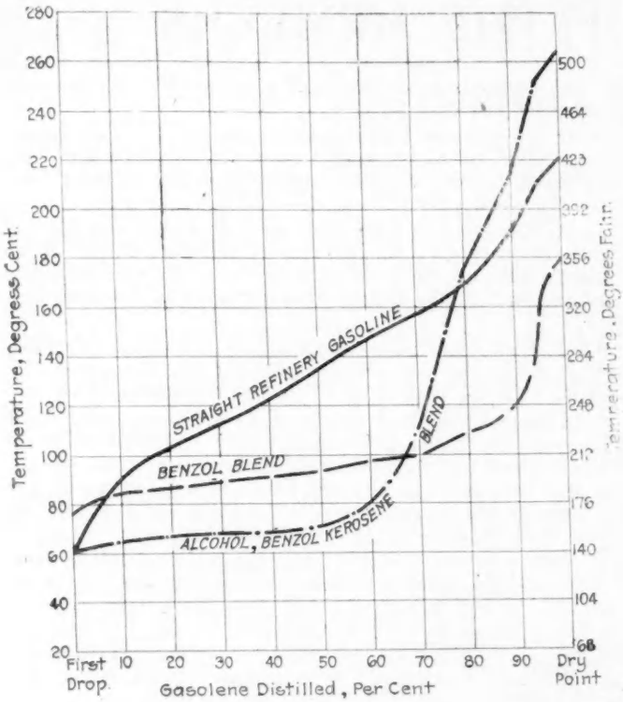


FIG. 16

cent. The real importance of coal-tar distillates as engine fuel occurs in districts adjacent to by-product coking operations, where a reasonable proportion of the local gasoline supply can be filled by coal-tar products. Benzol and other coal-tar distillates have certain disadvantages when compared with petroleum gasoline but have other very marked advantages, particularly in freedom from tendency to knock. An additional advantage lies in the fact that they have a considerably higher calorific value per unit volume and, as this is the basis on which the user buys, he gets more for his money than he would from petroleum gasoline. It is believed that coal-tar distillates, if properly used, are very satisfactory engine fuels and that there should be little difficulty in finding a market for the entire supply.

The Bureau's general feeling with regard to substitutes for gasoline is that they should receive all possible encouragement if produced and marketed in an intelligent and scientific manner. The nation certainly needs to increase its engine-fuel resources and no step in this direction should be hindered. It should be recognized, however, that petroleum gasoline is a type of fuel for which present equipment has been designed and for which it is eminently satisfactory. It is believed, therefore, that when striking claims are made for the superiority of any gasoline substitute, these should not be taken too seriously. The substitute fuel should be given a fair trial, but the user should be satisfied if it is as good as gasoline and should not expect it to be notably superior.

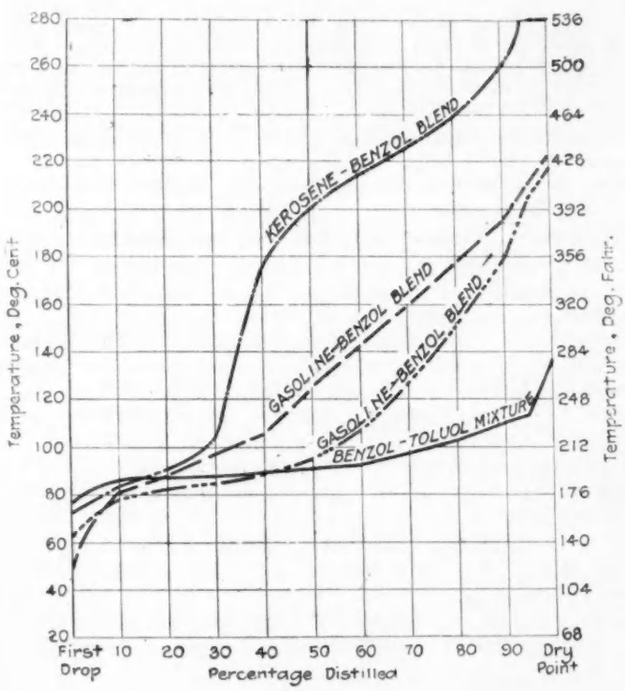


FIG. 17

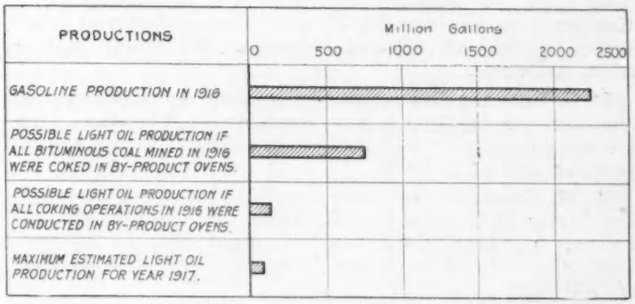

PRODUCTIONS
Million Gallons
0
500
1000
1500
2000
2500
GASOLINE PRODUCTION IN 1918
POSSIBLE LIGHT OIL PRODUCTION IF ALL BITUMINOUS COAL MINED IN 1918 WERE COKED IN BY-PRODUCT OVENS.
POSSIBLE LIGHT OIL PRODUCTION IF ALL COKING OPERATIONS IN 1918 WERE CONDUCTED IN BY-PRODUCT OVENS.
MAXIMUM ESTIMATED LIGHT OIL PRODUCTION FOR YEAR 1917.

FIG. 18

IMPORTANCE OF THE 5-TON TRACTOR

AN analysis of the total program for the production of tanks and tractors emphasizes the importance placed on the 5-ton tractor by the Ordnance Department. The following table gives the tractors ordered since April, 1917, and delivered or in course of construction on Sept. 23, 1919:

TYPE	NUMBER	PER CENT OF TOTAL NUMBER
5-ton Tractor	4,000	46.4
10-ton Tractor	2,800	32.5
6-ton Tank	950	11.0
20-ton Tractor	400	4.6
15-ton Tractor	267	3.1
30-ton Tank Mark VIII	100	1.2
2½-ton Tractor M 1918	87	1.0
3-ton Tank (experimental)	15	0.2
Total	8,619	100.0

Farmers Bulletin No. 1045 of the United States Department of Agriculture deals with the laying out of fields for tractor plowing. It has been prepared by the office of farm management and embodies descriptions of methods found best for nearly every type of farm in the country. A large portion of the information was compiled from reports of several hundred tractor owners. Among the subjects treated are the factors to be considered in making plans for tractor plowing and the methods by which the plows are lifted at the ends of the field. Nine different methods for laying out fields are presented, the text being supplemented by diagrams in each case. Other methods by which the plows are left in the ground in going across the ends are illustrated and described and schemes for irregular fields also receive attention.

PERSONAL NOTES OF THE MEMBERS

Charles L. Allmond has accepted a position as mechanical engineer with Smith, Hinchman & Grylls, Detroit, Mich. He was formerly a designer in the employ of the J. G. Brill Co., Philadelphia, Pa.

Horace A. Brown, Jr., who has been sales manager of the motor bearings division of the Hyatt Roller Bearing Co., Harrison, N. J., with headquarters at Detroit, Mich., has been appointed manager of the division to succeed B. G. Koether, who was recently elected vice-president and transferred to the main office at Harrison. Mr. Brown will remain at Detroit.

L. E. Butzman has accepted a position as assistant chief engineer with the Tool & Auto Products Co., Cleveland, Ohio. He was formerly in the employ of the Grant Motor Car Corporation, also of that city.

H. W. Crowhurst has been appointed sales engineer with the Carter Carburetor Co., St. Louis, Mo. During the war he was in charge of the road test department of the engineering division of the Motor Transport Corps with the rank of lieutenant.

James A. Edwards, who was formerly aeronautical engineer with the Hall-Scott Motor Car Co., San Francisco, Cal., has accepted a position as assistant engineer in the engine laboratory of the Associated Oil Co. of California, Oakland, Cal.

C. T. Fuller has resigned as management engineer with Frank B. Gilbreth, Inc., Providence, R. I., and has accepted a position as production manager with the Watson Products Corporation, Canastota, N. Y.

D. C. Goff has resigned as assistant sales manager of the Winther Motor Truck Co., Kenosha, Wis., and has accepted a position with the Leftwich Motor Co., Lynchburg, Va.

P. F. Hackethal has resigned as assistant chief engineer of the Templar Motors Corporation, Cleveland, Ohio, to accept the position of vice-president and chief engineer of the Paragon Motor Car Co., Connellsville, Pa.

R. O. Hendrickson, who was formerly chief engineer of the Wallis Tractor Co., Racine, Wis., has been appointed vice-president in charge of engineering of the J. I. Case Plow Works Co., also of that city, following the absorption of the Wallis company by the J. I. Case Plow Works.

K. W. Hooth has accepted a position as chief engineer with the Detroit Transmission Co., Detroit, Mich.

William L. Kaiser, who for the past 4 years has been sales engineer for the Sumter Division of the Splitdorf Electrical Co., Newark, N. J., has been appointed sales manager of the same division. He will continue to make his headquarters at the Chicago office of the company.

Romould Karasinski has been appointed assistant chief engineer of the Paragon Motor Car Co., Connellsville, Pa. He was formerly assistant chief engineer of the Cleveland Automobile Co., Cleveland, Ohio.

A. G. Kessler has been appointed general manager of the Buffalo plant of the Farrell Foundry & Machine Co., Inc., Waterbury, Conn. He was formerly with the General Ordnance Co., Derby, Conn.

A. M. Kimball has been appointed sales engineer for the Pierce-Governor Co., Inc., Anderson, Ind., in charge of the territory east of the Ohio-Indiana line. He was formerly chief engineer of the Fulton Motor Truck Co., Farmingdale, N. Y.

Albert E. Koeckert, who was formerly a mechanical draftsman at the Naval Gun Factory, Washington, has accepted a position with the Turbine Air Tool Co., Cleveland, Ohio.

B. G. Koether, who has been manager of the motor bearings division of the Hyatt Roller Bearings Co., Harrison, N. J., has been elected vice-president of the company. This promotion will transfer Mr. Koether from Detroit, where he has been for the last 10 years, to the main office of the company at Harrison, N. J.

The offices of the Leonard Tractor Co., of which Howard G. Leonard and H. M. Leonard are chief engineer and president and general manager respectively, have been removed from Jackson, Mich., to Gary, Ind.

A. H. Martin has been elected secretary and general manager of the International Steel Products Co., Hartford, Wis. He was formerly chief engineer of the Kissel Motor Car Co., also of that city.

George J. Mead has taken up the duties of quality manager with the Wright Aeronautical Corporation, New Brunswick, N. J. For the past year he has been engineer in charge of the power plant laboratory of the Air Service at Dayton, Ohio.

W. W. Mountain has been appointed vice-president of the Wright Roller Bearing Co., Philadelphia, Pa.

Walter C. Robbins has resigned as chief engineer of the Curtiss Aeroplane & Motor Corporation, Buffalo, N. Y., and is now practising consulting engineering in that city. He is specializing in internal-combustion engines, manufacturing, aeronautics and automobiles.

W. S. Roberts, who served with the Army in Europe for approximately 2 years, has accepted a position in the automobile equipment department of the Westinghouse Electric & Mfg. Co., East Pittsburgh, Pa., and is located at the company's New York City office.

S. R. Vande Water has accepted a position with the Kilbourne & Jacobs Mfg. Co., Columbus, Ohio. He was formerly an engineer in the employ of the Standard Steel Car Co., Pittsburgh, Pa.

William W. Zurlinden has had his name legally changed to William W. Linden.

The Velocity of Flame Propagation in Engine Cylinders

By DONALD MACKENZIE[1] (*Member*) AND R. K. HONAMAN[2] (*Non-Member*)

ANNUAL MEETING PAPER *Illustrated with* PHOTOGRAPHS AND DIAGRAMS

AMONG the important phenomena occurring in the gas engine which are of interest from the point of view of physicists and engineers alike, probably none has received so much attention as flame propagation with the accumulation of so few results directly applicable to operating conditions. Engine performance is very greatly influenced by the manner in which the

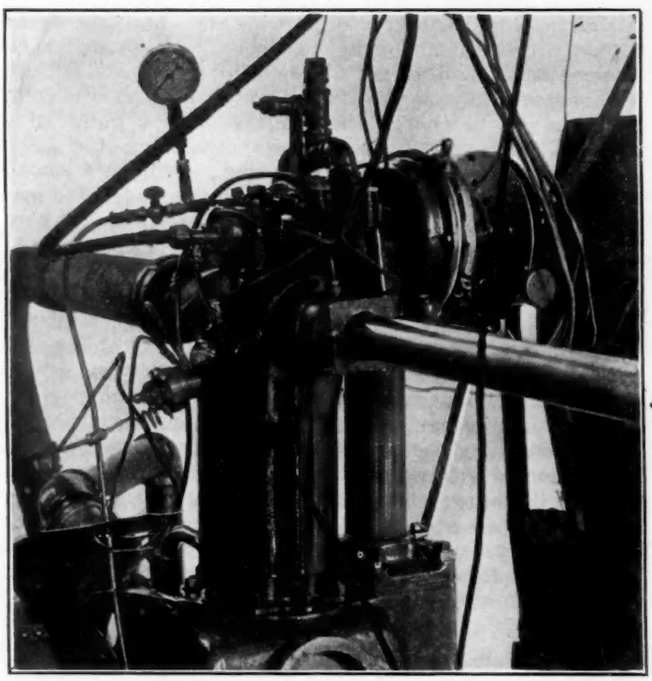


FIG. 1

pressure in the cylinder rises after ignition has taken place and burning progresses. In fact, the efficient use of the chemical energy of the gas depends among other things upon timing the ignition with relation to the position of the piston so that a maximum amount of the chemical energy is transformed into mechanical energy at the crankshaft. Moreover, the rate of rise of pressure is determined by the rate at which the flame spreads through the gaseous mixture.

This paper describes a method which has been devised for measuring the rate of flame propagation in gaseous mixtures and some experiments which have been made to coordinate the phenomena with the important factors entering into engine operation. The method used depends upon the fact that bodies at a high temperature ionize the space about them. These hot bodies may be inert substances or burning gases. For example, the flame in an engine cylinder constitutes such an ionizing agent. A spark-gap when bathed by the flame becomes a conducting medium and permits the discharge of a condenser, the terminals of which are connected to the electrodes of the spark-plug.

To make use of this phenomenon in the engine cylinder, experiments were made which showed that across a spark-gap in an atmosphere of compressed gas, such as that in an engine cylinder, a potential difference may be maintained which is just below the breakdown potential in the compressed gas before ignition, but which is sufficient to arc the gap after ignition has taken place and the flame has supplied ionization.

A cylinder-head in which the velocity is to be measured is arranged to take three spark-plugs Nos. 1, 2 and 3, all placed above the dead-center position of the piston. Plugs Nos. 2 and 3 are connected to a source of comparatively high electrical potential just before current for ignition is supplied to plug No. 1. At the high pressure existing in the cylinder during the latter part of the compression stroke, these gaps will not allow discharge. However, when the charge is ignited at plug No. 1, flame spreads out in all directions and

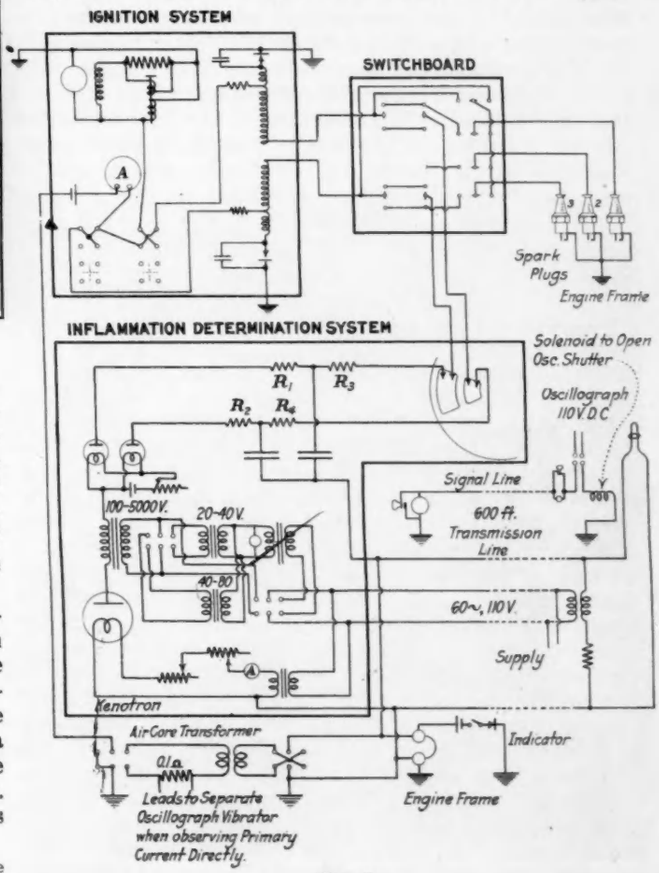


FIG. 2

[1]Assistant physicist, Bureau of Standards, Washington.
[2]Department of development and research, American Telephone & Telegraph Co., New York City.

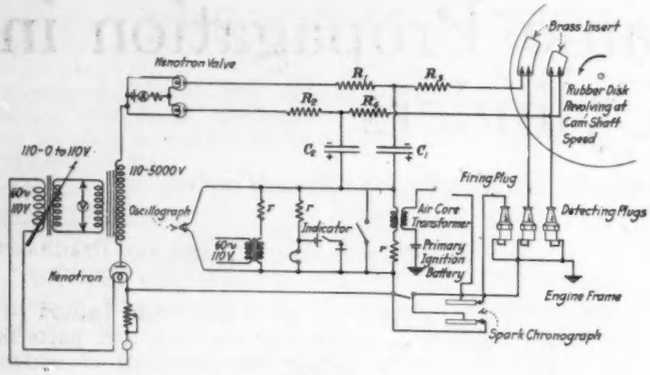


FIG. 3

ionizes the gas in gaps Nos. 2 and 3, permitting current from the source to discharge through them. By suitably connecting a recording device to register the instants when the igniting current is supplied to plug No. 1 and gaps Nos. 2 and 3 are ionized, and by determining therefrom the time required for flame to travel between plugs Nos. 1 and 2, and Nos. 1 and 3, the average velocity of the flame between these points is obtained.

How the Experiments Were Conducted

The experimental work was done on a one-cylinder Liberty engine, shown in Fig. 1. Fig. 2 is a complete wiring diagram of the electrical apparatus connected with the engine. In general it may be divided into three parts. The first is a standard Delco ignition system as supplied for use on Liberty engines. This includes two separate coils, a generator and a battery. The second is an arrangement of condensers and resistors that are used for determining the instant when the flame arrives at the observing spark-plugs. The third is a selector switchboard through which either of the two ignition coils and either of the two condensers can be connected to any one of the three spark-plugs, thus making the system entirely flexible and permitting any plug to be changed at will from an igniting plug to an observing plug.

The part of this set-up with which we are most concerned is the second part. Fig. 3 shows, simplified and on a larger scale, the system of condensers and resistors used for observing the instant of arrival of the flame. In this figure is shown a 100 to 5000-volt transformer which supplies the charge to condenser c_2 through a kenotron rectifying valve, high resistance r_2, and another rectifying valve v. The condenser c_1 is similarly charged through resistance r_1 and the valve associated with that circuit. The oscillograph vibrator, connected to the engine laboratory through a 600-ft. transmission line, is permanently in series with these two condensers. Furthermore, the condensers are connected through resistances r_3 and r_4 and through the contact-timing disk to spark-plugs Nos. 2 and 3. It will be noted that the oscillograph vibrator is also in series with this circuit. A battery ignition system supplies current to spark-plug No. 1. The instant of ignition is also recorded by the oscillograph vibrator by a small air-core transformer. The primary of this transformer is connected in series with the primary circuit of the ignition system and its secondary is in series with the oscillograph vibrator. When igniting current is supplied to spark-plug No. 1 by the opening of the primary contact-maker, the sudden decrease in current in the primary of the air-core transformer induces an electromotive force in its secondary and causes a deflection of the vibrator. After ignition, the flame reaches plugs Nos. 2 and 3 successively, discharging at the corresponding instants condensers c_1 and c_2 through resistances r_3 and r_4, and through the oscillograph vibrator, producing two successive deflections of the latter. The contact-disk, driven through gearing from the camshaft, serves the purpose of connecting the condensers to their proper spark-plugs only after the cylinder pressure has become sufficiently high to insure against breakdown before ionization by the flame. The phase of this disk is adjustable at will.

Resistances r_1 and r_2 are made about forty to fifty times larger than r_3 and r_4 so that very little current will be supplied from the alternating-current source to the spark-plug circuit; also, that the charging of condensers c_1 and c_2 shall be accomplished by a small current over a comparatively long time, and that the discharge through r_3 and r_4 shall be effected by a comparatively large current over a very short period, thus giving a sharp discharge signal on the oscillograph and charging signals that are broad and flat. The values are about 600,000 ohms for r_1 and r_2, and 15,000 ohms for r_3 and r_4. The condensers have a capacity of 0.0034 mf. It was found convenient to charge the condensers to a potential of about 3000 volts. Under these conditions, the initial discharge-current is about 0.1 amp., giving a 2-cm. deflection of the oscillograph vibrator. Moreover, the maximum charging current is not more than 0.005 amp., which gives a vibrator deflection of less than 1 mm.

The rectifying valves serve to isolate the condensers from each other during the discharge period. It was found that if r_1 and r_2 be directly connected, condenser No. 2 sometimes discharges through r_1, r_2 and r_3 into spark-plug No. 2, so that its potential is very greatly reduced before ionization occurs in spark-plug No. 3 and discharge can take place. These rectifying valves allow charging to take place freely but prevent any interaction of the two discharging circuits.

The discharge signals in the oscillograph were superimposed upon a 60-cycle wave from an alternating-current source, so that the oscillograph shutter might be kept open for several revolutions of the drum without having the observations overlap. The effect of this will be shown in a later figure.

Recording the Results Photographically

This same set-up has also been arranged to show on the films the instant when maximum pressure is reached in the cylinder. The apparatus for accomplishing this is also shown in Fig 3, which is a skeleton diagram equivalent to Fig. 2. The contact-maker, not shown in Fig. 3, is a diaphragm and pin. One side of the diaphragm is exposed to the pressure of the engine gas and the other side to a controllable static pressure. The static pressure is raised so that at no time during the engine cycle does the diaphragm make contact. It is then gradually lowered until contact is made during one short interval in each cycle, as shown by a click in the telephone receiver. This is obviously during the instant of maximum pressure. Current from a small battery, not shown in Fig. 3, then flows during this short interval, adding another excursion of the oscillograph vibrator. This permits a determination of the instant of maximum pressure in relation to the piston position. A complete description of the indicator is given in another paper presented at this meeting.

Fig. 4 shows the type of record that was obtained. The first excursion of the vibrator occurred when the

ignition current was supplied through the air-core transformer to spark-plug No. 1, and as spark-gaps Nos. 2 and 3 were ionized and condensers c_1 and c_2 discharged through them, the second and the third excursions were made. The frequency of the alternating-current wave is known and consequently the time between ignition and each of the excursions can be determined. Fig. 4 does not show the type of record obtained for the maximum-pressure determination, but the apparatus can be designed so that an additional record is obtained at the instant of maximum pressure.

The three waves were traced during three successive revolutions of the drum and the speed was adjusted so that the observed displacement was obtained. The graph drum and is graduated to show the position of the sight-line in degrees from a given reference point. The sight-line is first set on the spark made by the ignition system and is then turned to the spark caused by current discharging through one of the spark-plugs; the angular distance through which it is moved is then noted. Calling this angle θ and the distance between the spark-plugs in meters d, the velocity of flame propagation can be determined directly from the formula:

$$v = \frac{6nd}{\theta - \alpha}, \text{ where}$$

n = the speed in r.p.m.; and
α = 18 deg., the fixed angular separation of the ignition and condenser gaps.

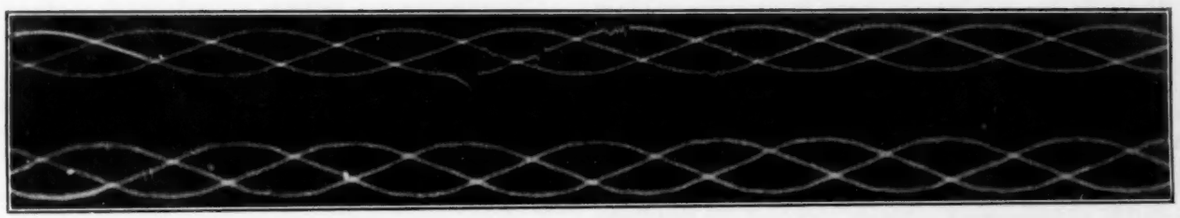


FIG. 4

first excursion shows the instant that current is supplied to the igniting spark-plug; the second and third excursions indicate the succeeding instants when ionization takes place in the observing plugs. It might be noted that the ignition signal is made up of several oscillations; these are probably caused by the fact that the rate of change of the primary current of the ignition system is fluctuating, giving variations in the voltage in the secondary of the air-core transformer.

To release the oscillograph for other investigations, an instrument has been designed to replace it for this work. The new recording device may be known as a spark chronograph. This consists of a cylinder of electrical insulating material containing brass-ring inserts, about which are rotated metallic sparking points radially placed. If a sheet of carbon paper is placed on the cylinder a small spark-gap is formed between the sparking points and the metal rings, and the paper is punctured. Fig. 3 shows diagrammatically the location of such an instrument.

THE SPARK CHRONOGRAPH

Fig. 5 shows the spark chronograph that will replace the oscillograph. In this way there is obtained an estimate of the variation in flame propagation from cycle to cycle. The spark is easily visible and should occur at the same position on the drum, when the sparking points are rotated synchronously with the engine. If the spark moves about from point to point, it is evident that the instant of ionization for that particular spark-plug is changing from cycle to cycle. In obtaining a record, the spark chronograph must be left running as long as desired and the discharge current will then puncture a new hole for each engine cycle. The center of this group of holes is then the average position for the instant of ionization in the corresponding plug, taken throughout the operating period of the chronograph. The ignition current is also permitted to make a record on the sheet, under another spark-plug.

The spark chronograph can also give a visual indication of the velocity of flame propagation. A disk with sight-vanes is rotatable about the axis of the chrono-

The results so far obtained show that the fundamental idea involved in this method is sound, namely, that a voltage sufficient to arc a gap ionized by the ignition flame is not large enough to break down the compressed gas. To test this in each case we have determined that the voltage used on the plugs would not interfere with engine operation. Measurements obtained from different records indicate that the velocity of flame propagation varies widely with different engine conditions and also that the velocity of propagation increases rapidly as the flame travels away from the point of ignition.

The value of the method already described lies in the

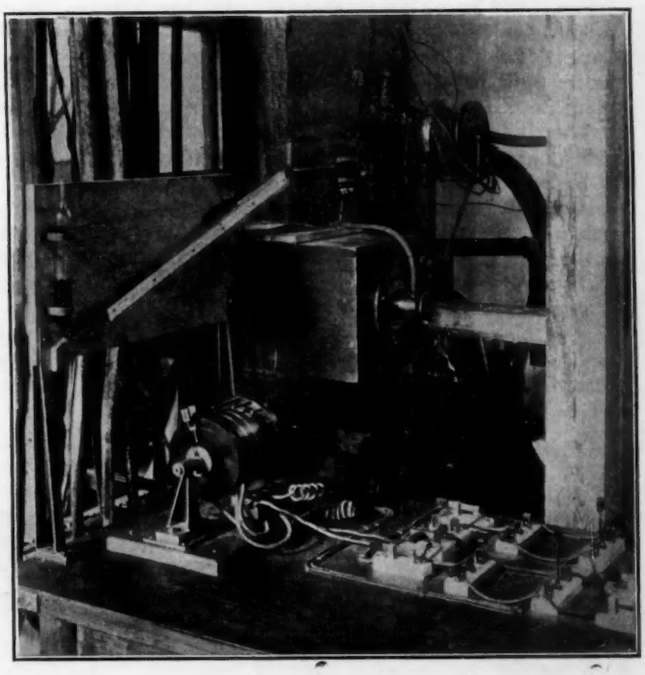


FIG. 5

fact that by its use the velocity of flame propagation can be studied in an engine while it is actually in operation. The results obtained are in marked contrast to those derived from experiments with explosive mixtures in closed vessels, but they furnish a safer basis for comparisons of fuels and for the designing of cylinder-heads.

In any particular engine, various fuels and different ignition systems may be studied and their relative merits indicated. Location of spark-plugs and shape of combustion space have an influence upon the time taken for complete inflammation and hence upon the interval between ignition and the attainment of maximum pressure. If several designs of cylinder-head are available, the method can be used to determine their relative merits.

Studying Turbulence and Knocking

Moreover, some conclusions can be drawn as to turbulence in the mixture, its magnitude and its effect upon horsepower output. By successively changing the functions of the three plugs, using No. 1 as the firing plug and Nos. 2 and 3 as observing plugs, then by changing the ignition to plug No. 2 and finally to plug No. 3, it is possible to compare the times occupied by the flame in traveling in two directions over each of three different paths. If the time interval from plug No. 1 to plug No. 3 is greater than that from plug No. 3 to plug No. 1, it might be concluded that a current of gas from plug No. 3 toward plug No. 1 gives a velocity to the flame in addition to its normal velocity of propagation.

Experiments have shown turbulence effects with velocities from 1 to 4 m. per sec. When the engine is fired at one plug alone and this is the one from which the flame-spread is accelerated by turbulence, measureably greater horsepower is developed, reaching in some cases an increase of 5 per cent over the output when the ignition takes place at an opposite plug.

During the development of the method of measurement mentioned, the output of the engine was absorbed by a fan brake. Under these circumstances close control of the speed is impossible, nor were there proper means for determining and holding a constant mixture ratio. The engine is at present coupled to a dynamometer and provision is made for obtaining more complete data.

The phenomena of knocking are not yet fully investigated, but it was observed that when violent knocking is heard the oscillograph records abnormally short time-intervals. If the spark is then retarded until knocking ceases, the time-interval recorded returns to its normal value. It is yet too early to predict how much information regarding knocking can be derived from studies of the velocity of flame propagation.

It is planned to compare different fuels in the engine and to study the variations in flame velocity with a change in the mixture ratio, compression pressure, throttling, spark advance and speed. Comparisons of various designs of cylinder-head are also hoped for.

The results of several hundred observations are given. The data were collected under these conditions:

Standard "X" gasoline
Mixture ratio, about 12 parts air to 1 of fuel
Speed, about 1400 r.p.m.
Full throttle and 30-deg. spark advance
Carbureter, Tice experimental

When results in opposite directions over the same path are averaged to eliminate the effect of turbulence, they appear as follows:

x=Distance, cm.	t=Time, sec.	Average Velocity=$\frac{x}{t}$ m. per sec.	ft. per sec.
2.8	0.00434	6.44	21.13
9.5	0.00872	10.87	35.66
11.8	0.00937	12.58	41.27

A later series of observations over the second of these distances, made with a Claudel carbureter, gave the result x=9.5 cm. and t=0.00858 sec. or practically the same as the value tabulated. These values of x and t satisfy within 1 per cent the empirical formula.

$$\frac{x}{t} = ax + b$$

From this we obtain

$$x = \frac{bt}{1-at} \text{ and } \frac{dx}{dt} = v = \frac{b}{(1-at)^2} \text{ or } b\ (\frac{a}{b}x + 1)^2$$

The flame velocity is seen to be low in comparison with the velocity of the air inflow, about 200 ft. per sec. The piston speed, 1600 ft. per min., is intermediate between the low and the high flame velocities observed, 1270 and 2480 ft. per min. respectively.

It is to be observed that this expression for the velocity of flame propagation involves two constants a and b, the ratio of which appears in the parenthesis as the coefficient of the distance from the firing plug. It may be conjectured that the constant b is a property of the fuel mixture used, while the appearance of x shows the effect of confinement of the explosion in the cylinder-head. This effect should depend upon both the fuel and the shape of the combustion space. It may develop that a is a coefficient whose value depends on the geometry of the explosion chamber. It is expected that further data will yield a criticism of this formula and perhaps a physical interpretation of it.

Writing v=velocity of flame propagation in centimeters per second, the constants have the value a=68.6, b=450; so that

$$v = \frac{450}{(1-68.6t)^2}, \text{ or } 450\ (0.153\ x + 1)^2$$

Inspection of the formula suggests at first that v increases without limit, but it must be remembered that t has a limit, namely, the instant when inflammation is complete and flame spread ceases.

Dilution of Engine Lubricants by Fuel

By GUSTAVE A. KRAMER[1] (*Non-Member*)

ANNUAL MEETING PAPER *Illustrated with* CHARTS

PARALLEL with the decline in the volatility of commercial gasoline, complaints have appeared with increasing frequency of engine lubrication troubles resulting from the dilution of the lubricating oil in engine crankcases. The trouble is particularly pronounced in tractor engines where economy demands the use of the cheapest possible grade of fuel.

How much of the trouble reported is really caused by dilution of the lubricant would be difficult to trace. Nevertheless it must be admitted that the problem does exist and that it is becoming more serious with the steady decrease in volatility of internal-combustion engine fuel. Inasmuch as this decrease is due to an increased demand, no relief through an increase in volatility of the fuel is in sight. The situation must then be squarely faced by the engineers of the country and attacked from every direction that promises a solution of the problem.

The most satisfactory method of handling investigations of this kind is usually that of elimination; the finding of answers to the simple questions which form a part of the more complex problem. The Bureau of Standards is attempting to throw light on a few phases of the subject. The work described was done as a part of the general investigation of lubricants and lubrication now under way and was naturally confined to the influence that the properties of the available lubricating oils have on the phenomena of crankcase dilution. The properties of the oils must to a certain extent be regarded as rigidly fixed, the oils being products of refining from available crude oils as are also the fuels under consideration. There exists a wide range of viscosities from which to choose, but the possible range for a given type of engine is comparatively narrow and still narrower for any individual engine. From our present knowledge, there is little hope of finding an oil whose viscosity will not be affected by the admixture of gasoline and kerosene. Even castor-oil, the only commercially available lubricant that at ordinary temperatures does not mix with gasoline and kerosene in all proportions, loses this property at engine temperatures. Experiments made by the Bureau of Standards on a commercial sample have shown that it dissolves an equal volume of kerosene at 35 deg. cent., (95 deg. fahr.), and an equal volume of gasoline at 24 deg. cent., (75 deg. fahr.). In fact, above 43 deg. cent., (110 deg. fahr.), either gasoline or kerosene will dissolve in castor-oil in any proportion, and the reduction in the viscosity of castor-oil is probably no less for the same dilution than that of an equally viscous petroleum oil. (See Fig. 2). Since an oil is usually above 40 deg. cent., (104 deg. fahr.), the major portion of the time that an engine is in operation, it is evident that castor-oil offers no solution of the difficulty. Unless some unforeseen discovery should enable us to produce economically a lubricant not miscible with our present engine fuels, we must expect to lubricate our engines with oils which steadily decrease in viscosity whenever a part of the fuel can find its way into the oil. To substitute oils of higher viscosities so that even considerable dilution could be permitted would only mean postponing and not preventing the trouble.

TWO POSSIBLE SOLUTIONS OF THE PROBLEM

This being the case, two logical ways remain for overcoming the difficulties. The first and most satisfactory is to prevent dilution of the oil. The second is to separate the fuel from the oil, thereby restoring the oil to its original viscosity as nearly as is possible.

To prevent absorption of the fuel by the oil is a problem of engine design. Better atomization and vaporization would undoubtedly aid. The cross-head-type engine, whose cylinders are relieved of all side-pressure and therefore require only sparing lubrication, might afford a step in the right direction. But the elaboration of these possibilities is beyond the scope of this paper.

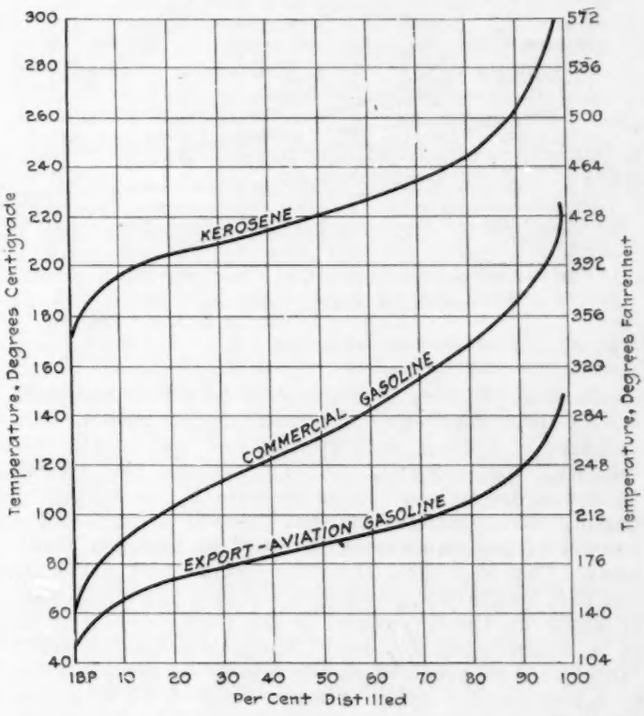


FIG. 1

When considering oil dilution by fuel, it should be remembered that, during the suction and compression strokes, the lubricant on the cylinder-walls is in contact with the air-fuel mixture. This oil may absorb part of the fuel and then drip from the cylinder-walls into the crankcase. Moreover, even the best fitting pistons and rings allow some of the cylinder charge to leak into the crankcase during compression.

Experiments were therefore undertaken to determine to what extent the absorption of fuel vapors can occur at average engine temperatures. A simple apparatus was set up and arranged so that a measured volume of a

[1] Chemical engineer, Bureau of Standards, Washington.

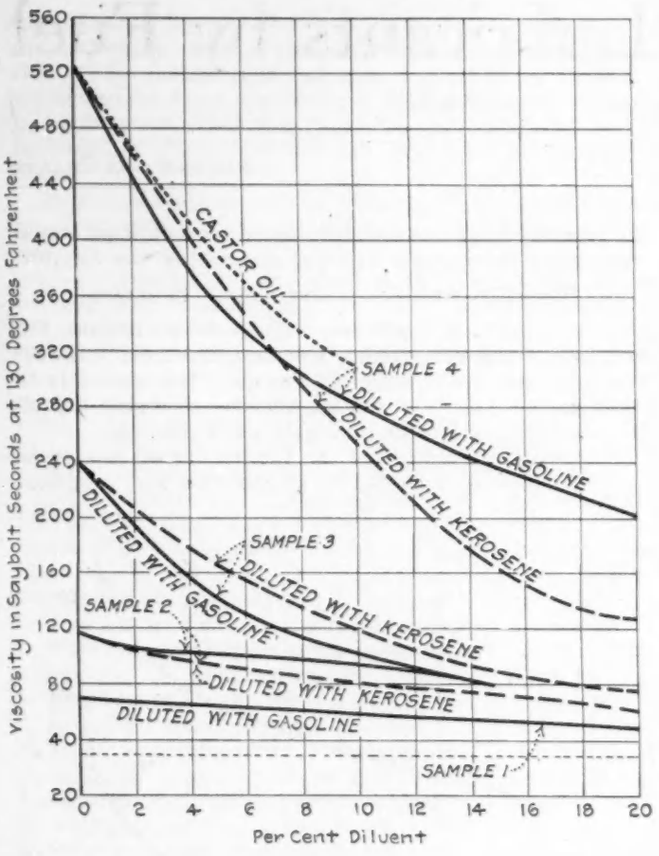


FIG. 2

uniform gasoline-air mixture could be bubbled through the oil under examination. A steam bath permitted temperature control of both the oil and the gasoline-air mixture. The amount of vapors absorbed during a given time was determined by the increase in the weight of the oil. The results show that there is a limit to the amount of gasoline vapors absorbed under these conditions. That this limit is rather definite was shown by the fact that after a certain gain in weight had been observed and the oil replaced in the system, the drawing of a second equal volume of the gasoline-air mixture through the oil produced no increase in weight. The use of a considerably richer mixture resulted, as would be expected, in a greater increase in weight. Repeating the first experiment with oil to which a quantity of gasoline had been added, a decrease in oil weight resulted such that at the end of this experiment it was approximately the same as at the completion of the first experiment. These experiments thus indicate that a point of equilibrium is soon reached, at which the gasoline evaporates from the oil at the same rate as that at which the oil absorbs it. The amount of gasoline in the oil when this point is reached is not large, ranging from 0.4 to 2 per cent with different oils and various gasoline-air ratios. It should be noted in connection with this experiment that a high grade of gasoline was used throughout so that it entirely evaporated in the air-current before reaching the oil.

Although the method just described was an approximation of the conditions to which a lubricant is subjected in an engine as regards gasoline-air mixtures, it was thought that apparatus based on a different principle, that of partial vapor-pressures, would be more satisfactory for further work. Such apparatus has been built and sufficient work has been done to indicate that the point of equilibrium already mentioned, while varying with different oils, is in no case greater than the maximum given. From present indications it must consequently be concluded that this source of dilution, the absorption of fuel vapors at engine temperatures, is negligible.

Cracking Process in Engines

There is another method by which the viscosity of oils may be reduced that is the topic of much discussion at present. The assertion is frequently made that there is no need of explaining the observed reduction in viscosity of lubricants by their dilution with unburned fuel, because a process similar to that used in the manufacture of gasoline from the heavier hydrocarbons, called "cracking" by refiners, is continuously in progress on the overheated metallic surfaces of the engine. I have never heard any oil-refiner make or sustain this statement, but the opinion of the Bureau of Standards on this subject has been sought so frequently that a few remarks seem warranted.

The results from a series of oil tests made by the Bureau of Standards, in an aircraft engine operating continuously for periods of from 5 to 11 hr. at nearly full load, showed no indication of dilution due to cracking with the oils used, these being representative of the refiners' products from the typical crudes available in this country. In these tests the viscosities of all the oils used increased steadily during the runs. The amount of light fractions, those distilling below 232 deg. cent., (450 deg. fahr.), found in the crankcase oil, was at no time during the test greater than 1.5 per cent by volume; and an examination of these light fractions showed nearly the same percentage of unsaturated compounds as the original fuel. Had there been any dilution due to cracking, this would have been indicated by the presence of a larger proportion of unsaturated compounds.

It is not claimed here that some destructive distillation of the oil does not occur in engines. It is probable that

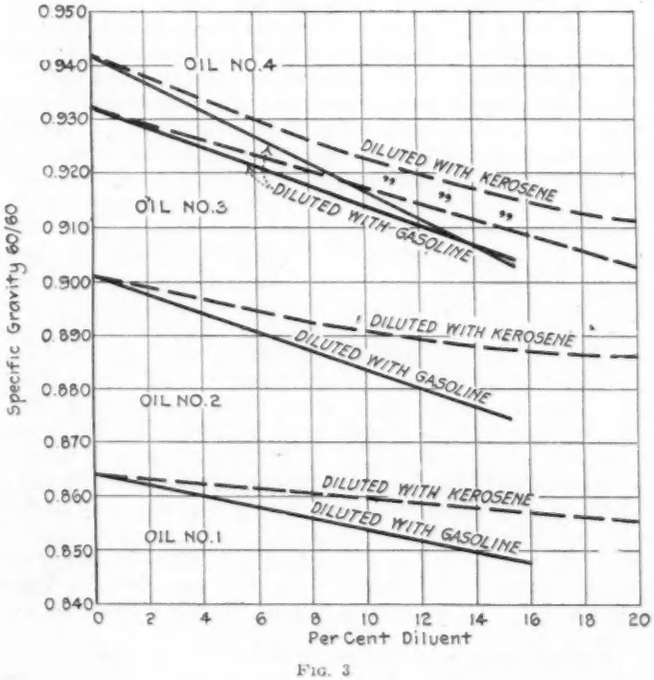


FIG. 3

whenever a few drops of oil come into contact with an engine part hot enough to cause cracking, the light products formed are spontaneously evaporated by the heat of the part. True cracking of hydrocarbons occurs, to a considerable extent, only at a much higher temperature than is generally imagined and seldom below 300 deg. cent., (572 deg. fahr.). Tests on an airplane engine were selected because of the high temperatures to which the oils are subjected in this type. The temperatures in tractor engines are not much lower, but in neither type, if well designed, do enough parts ever attain sufficiently high temperatures to give any considerable amount of cracked products.

To prove or disprove conclusively whether spontaneous decomposition of an oil can occur at the temperatures

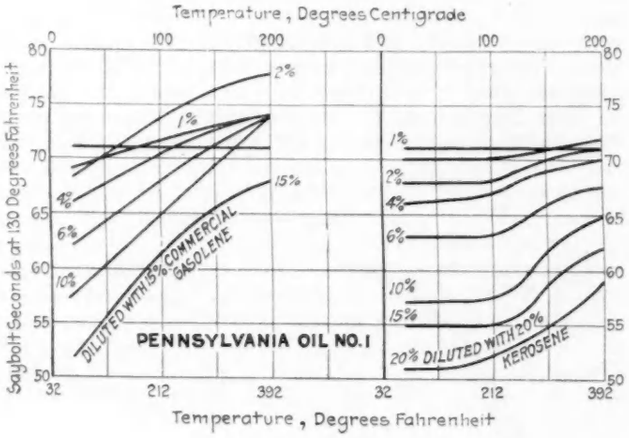

Temperature, Degrees Centigrade
Saybolt Seconds at 130 Degrees Fahrenheit
DILUTED WITH 15% COMMERCIAL GASOLENE
PENNSYLVANIA OIL NO.1
20% DILUTED WITH 20% KEROSENE
Temperature, Degrees Fahrenheit

FIG. 4

prevailing in internal-combustion engines, requires the determination of the cracking temperature of that oil. The Bureau of Standards has devised a simple apparatus for this purpose, although this is still in process of development. Until such apparatus is proved satisfactory, or until further information is obtained, judgment as to cracking of oils in engines must rest on the fact that no dilution due to decomposition has been found to any considerable extent in used lubricating oils.

POSSIBILITY OF IMMEDIATE RELIEF

Even if an engine should be developed within a short time that conformed to the requirements necessary to prevent oil dilution, relief would reach the owner only as the engines in use were replaced by the new design, and this would be a matter of several years. For the intervening period, the second means suggested for preventing lubrication trouble from fuel dilution may be found applicable. This is to prevent failure of lubrication by removing a sufficient portion of the fuel which has entered into the oil. The simplest practical procedure for this would appear to be some process of distillation, carried on at temperatures high enough to evaporate the major portion of the absorbed fuel, yet below those at which oxidation of the oil would occur to a serious extent.

Before attempting to apply the great fund of available information on distillation methods to the solution of this problem, data are necessary on three questions relative to lubricating oils:

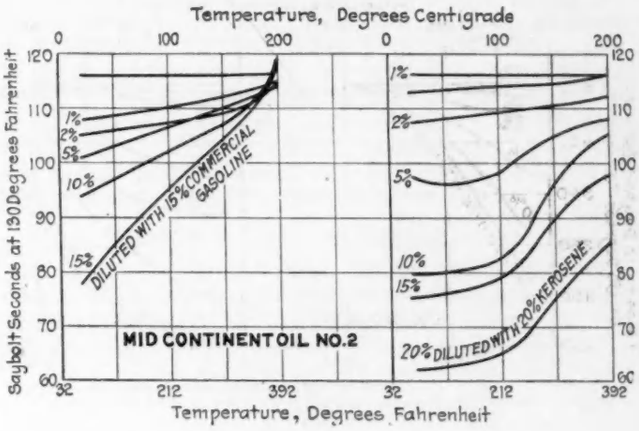

Temperature, Degrees Centigrade
Saybolt Seconds at 130 Degrees Fahrenheit
DILUTED WITH 15% COMMERCIAL GASOLINE
MID CONTINENT OIL NO.2
20% DILUTED WITH 20% KEROSENE
Temperature, Degrees Fahrenheit

FIG. 5

(1) What is the effect on the viscosity of lubricating oils of their dilution with varying percentages of fuels? The allowable percentage of fuel dilution can be determined when the permissible viscosity-range of the lubricant for the engine is known

(2) What is the maximum temperature to which lubricating oils can be raised without serious deterioration loss or danger when: (*a*) exposed to the air and (*b*) exposed to their own vapor? No definite answer can be given to this question at present although, as has been stated, it is not probable that serious cracking of properly refined lubricants will occur at temperatures lower than 300 deg. cent., (572 deg. fahr.). Oxidation will occur even at lower temperatures, but its extent will depend upon the exposed surface, the amount of air and the time of heating

(3) What are the maximum temperatures to which oils must be heated to remove the necessary proportion of the fuels which they contain? The maximum temperature which it may be necessary to reach to remove a given fuel can be judged from the upper distillation temperatures of the fuel, but experiments described later indicate that much lower temperatures than this will suffice.

EFFECTS OF FUEL ADMIXTURE ON VISCOSITY

A series of experiments to determine the effect of dilution by fuels on the viscosities of lubricants has been carried out with four oils of different origin, representative of the principal crude oils of the country and having viscosities usually encountered in lubricants for internal-combustion engines. These oils were diluted

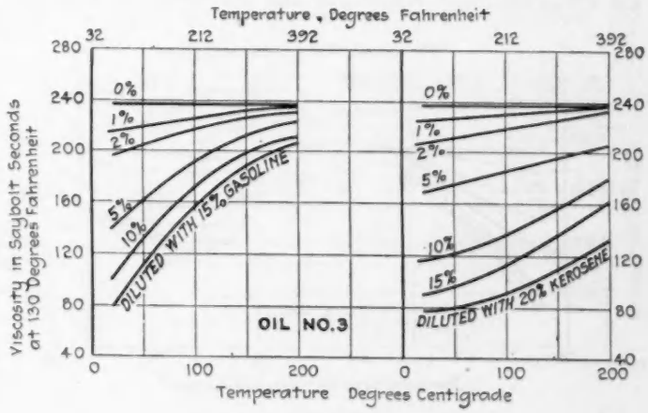

Temperature, Degrees Fahrenheit
Viscosity in Saybolt Seconds at 130 Degrees Fahrenheit
DILUTED WITH 15% GASOLINE
OIL NO.3
DILUTED WITH 20% KEROSENE
Temperature Degrees Centigrade

FIG. 6

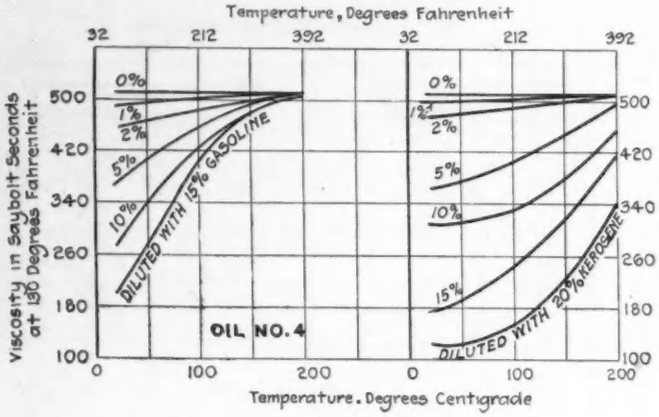

Temperature, Degrees Fahrenheit
32 212 392 32 212 392
Viscosity in Saybolt Seconds at 130 Degrees Fahrenheit
500 420 340 260 180 100
0% 1% 2% 5% 10% 15%
DILUTED WITH 15% GASOLINE
DILUTED WITH 20% KEROSENE
OIL NO. 4
0 100 200 0 100 200
Temperature. Degrees Centigrade

FIG. 7

with two fuels in varying percentages. The fuels used were:

(1) An average grade of fuel gasoline
(2) A sample of commercial kerosene

Distillation curves of these fuels are given in Fig. 1, as determined by the method for distillation of gasoline developed by the Bureau of Mines. The viscosity of each oil was first determined undiluted and then when diluted with different percentages by volume of the two fuels.

The results of this series of experiments are given in Figs. 2 and 3. Fig. 2 shows the change in viscosity of the four different oils, expressed in Saybolt-seconds, resulting from their dilution with varying quantities up to 20 per cent by volume of commercial gasoline and kerosene. The results of a few experiments with aviation gasoline are similar in their general characteristics to those of commercial gasoline and are omitted to avoid confusion. The values of the viscosities of the gasoline and kerosene used as diluents are about 29 and 31 Saybolt-seconds, respectively.

For the same dilution, the decrease in viscosity of the more viscous oils is much greater than that of the lighter ones. Where nearly the same viscosity-range is permissible with a heavy and a light oil, a higher precentage of dilution would be permitted with the lighter oil. Fig. 3 shows the change in specific gravity of the same oils when diluted with different percentages of fuel gasoline and kerosene.

Data such as those given would show what percentage of dilution could be permitted when the viscosities required for any given service were known. The numerical value of these viscosities and the permissible range

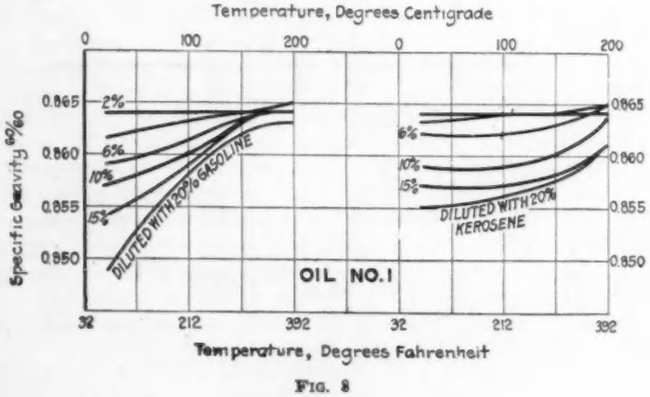

Temperature, Degrees Centigrade
0 100 200 0 100 200
Specific Gravity 60/60
0.865 0.860 0.855 0.850
2% 6% 10% 15%
DILUTED WITH 20% GASOLINE
DILUTED WITH 20% KEROSENE
OIL NO. 1
32 212 392 32 212 392
Temperature, Degrees Fahrenheit

FIG. 8

of their variation must be determined for individual cases. The data given are, however, suggestive of what can be expected with existing oils.

SEPARATION BY DISTILLATION

When considering distillation methods for the removal of the fuel from the oil, the simplest method is the heating of the liquid to be treated in an open dish exposed to the air. It was felt that data showing the results of the use of this very simple process, when applied to oil diluted with fuel, would serve to indicate the degree of refinement necessary in distillation methods to be used for the removal of the fuel from oil, and the maximum temperatures which it would be necessary to reach.

With this in view, the samples of the four different oils previously diluted with varying percentages of fuels were used. In carrying out the experiments each sample of oil was heated to 100 deg. cent., (212 deg. fahr.), held at this temperature for 5 min. and then allowed to cool. Its viscosity and specific gravity were then measured. The same sample was then heated to 150 deg. cent., (302 deg. fahr.), held there for 5 min. and again allowed to cool. Its viscosity and specific gravity were again measured. Finally this operation was repeated with the

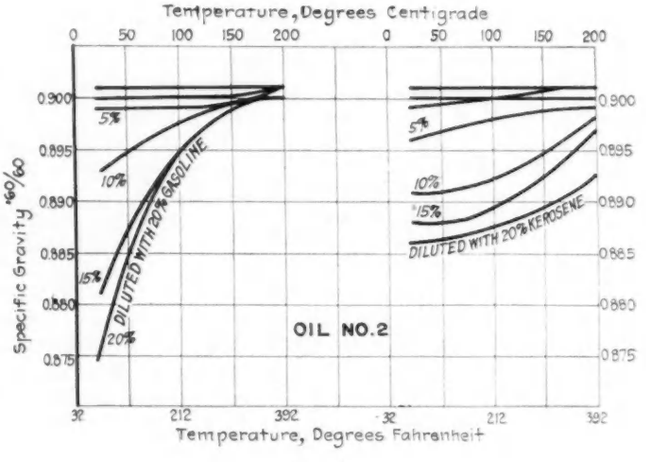

Temperature, Degrees Centigrade
0 50 100 150 200 0 50 100 150 200
Specific Gravity 60/60
0.900 0.895 0.890 0.885 0.880 0.875
5% 10% 15% 20%
DILUTED WITH 20% GASOLINE
DILUTED WITH 20% KEROSENE
OIL NO. 2
32 212 392 32 212 392
Temperature, Degrees Fahrenheit

FIG. 9

same sample held at a temperature of 200 deg. cent., (392 deg. fahr.), for 5 min. Figs. 4 to 11 show the results of these experiments. In Figs. 4 to 7 the change in viscosity of the oil-fuel mixtures after heating successively to 100 deg. cent., (212 deg. fahr.); 150 deg. cent., (302 deg. fahr.); and 200 deg. cent., (392 deg. fahr.), is given.

After heating to 150 deg. cent., (302 deg. fahr.), the oils diluted with commercial gasoline were restored to what would seem to be a safe viscosity, considering the original value for the oil. The results with aviation gasoline which are not shown on the curves are very similar. Practically all the gasoline had been removed at 150 deg. cent., (302 deg. fahr.), and the greater proportion at 100 deg. cent., (212 deg. fahr.). With kerosene the results are not so encouraging. In the attempt to remove the kerosene, heating to 100 deg. cent., (212 deg. fahr.), had practically no effect in the case of the two lighter oils and with the heavier oils much less effect than the heating to 150 deg. cent., (302 deg. fahr.). An examination of the distillation curves of the fuels will assist in explaining this result by showing the effect of the volatility of a fuel on the temperature to which its mixtures with oil must be heated to effect its removal.

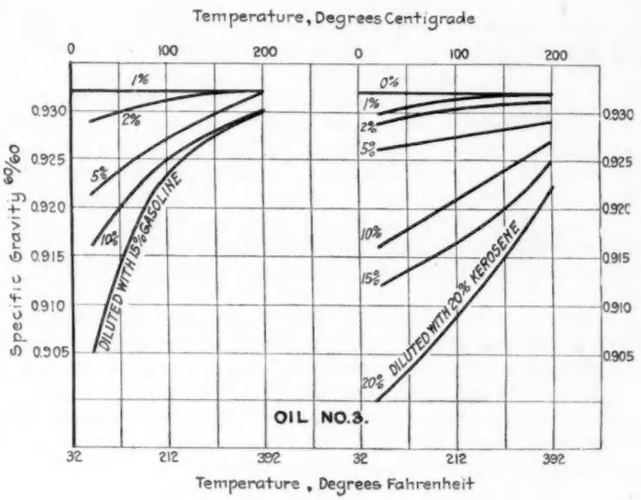


FIG. 10

When heated to 100 deg. cent., (212 deg. fahr.), in the presence of its own vapor, about 20 per cent of commercial gasoline will evaporate; when heated to 150 deg. cent., (302 deg. fahr.), about 67 per cent evaporates. When other vapors are present, as is the case in the simple distillation used to remove the fuels from the oil, the partial pressure of the fuel is lowered by its dilution in the air and at the same temperature a greater volume of the fuel will evaporate than when its vapor pressure is higher. The same explanation will show why little, if any, of the kerosene is removed when the diluted oil is heated to 100 deg. cent., (212 deg. fahr.), and an appreciable amount at 150 deg. cent., (302 deg. fahr.), and 200 deg. cent., (392 deg. fahr.), although when heated in the presence of its own vapor only 20 per cent of the kerosene vaporizes at 200 deg. cent., (392 deg. fahr.), and but a very small amount at 150 deg. cent., (302 deg. fahr.), and 100 deg. cent., (212 deg. fahr.).

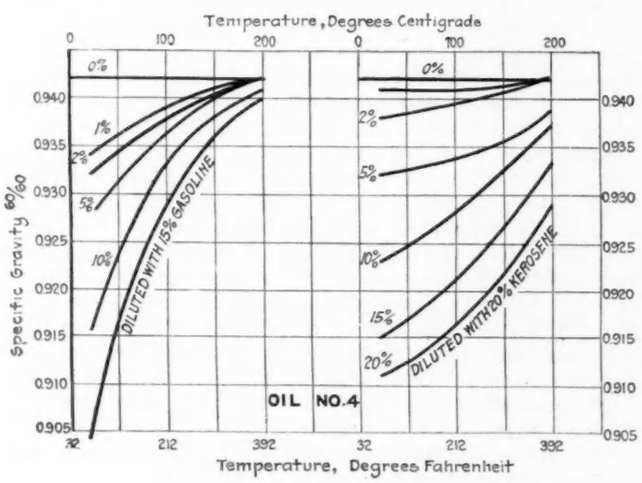


FIG. 11

The values for the specific gravities of the diluted mixtures are given in Figs. 8 to 11, to show that specific gravity can be used to estimate roughly the changes in viscosity resulting from fuel dilution. It should, however, be carefully borne in mind that, in general, specific gravities and viscosities are not proportional.

The results of the experiments with the simplest method of distillation show that there is no necessity for highly developed distillation methods when the present-day commercial engine-gasoline or a more volatile fuel is to be removed from lubricating oils, provided, of course, that the composition of the oil will permit its heating to 150 deg. cent., (302 deg. fahr.), or 200 deg. cent., (392 deg. fahr.). There is, however, a need for the use of better methods for the removal of kerosene and similar less volatile fuels from the oils.

Effects of Time and Exposed Area

With this in view, several further experiments were carried out to indicate the possible improvements through simple changes in the method of distillation. In this

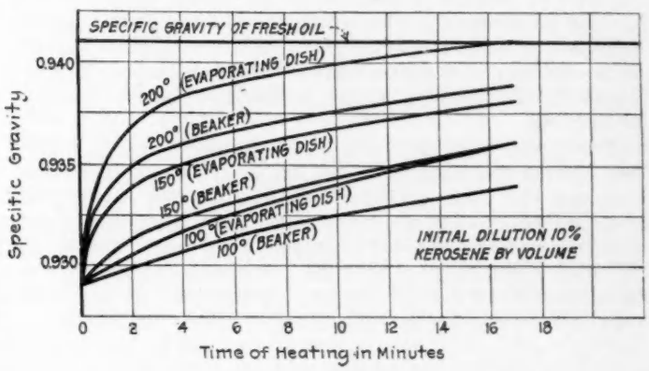


FIG. 12

series the effect of changes in the time of heating and in the area of free surface exposed, on the removal of fuels from oil was studied. An engine oil of medium viscosity was diluted with various percentages by volume of the same two fuels previously used, engine gasoline and kerosene. A sample of about 100 c.c. of the diluted oil was heated in a 200-c.c. beaker to 100 deg. cent., (212 deg. fahr.), and held at this temperature for 2 min. It was then allowed to cool and its specific gravity was measured. This procedure was then repeated, but for different lengths of time; for 5 min. in the first instance and 10 min. in the second. This series was then repeated with a fresh sample of oil at 150 deg. cent., (302 deg. fahr.), instead of 100 deg. cent., (212 deg. fahr.), and again with another sample of oil at 200 deg. cent., (392 deg. fahr.).

To show the effect on the removal of the fuel of a change in the area of the free surface of the oil, the series of operations just described was repeated in an evaporating dish giving about four times the area of exposed oil-surface. Another method of increasing the

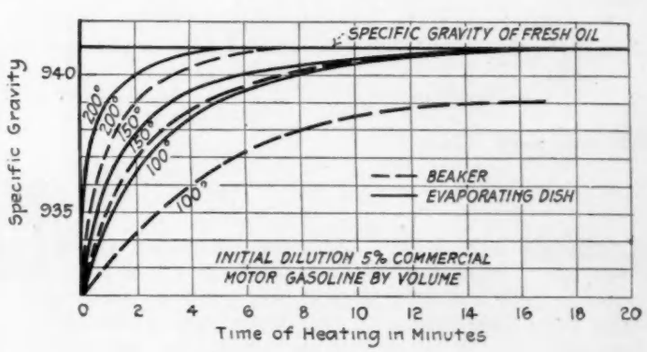


FIG. 13

effective area of oil surface for evaporation would consist in passing a stream of air through a comparatively deep layer of oil.

The results shown in Figs. 12 and 13 are typical. In these figures specific gravity is used to indicate the extent to which the fuel had been removed, in the absence of a complete series of viscosity readings that could not be made in the time available. The validity of this use of specific gravity as an index of viscosity was checked, however, by viscosity measurements.

The results given in Fig. 12 are with 10 per cent of kerosene as a diluent. They show clearly that the time of heating and the exposed area of oil surface have very marked effects on the removal of fuel from the oil. The results of the experiments made, using greater and smaller dilutions with kerosene, showed the same marked increase in amount of fuel removed with increase in surface exposed. They also showed that the larger the fuel dilutions, the more pronounced was the effect of the time of heating.

For commercial gasoline, it is shown in Fig. 13 that for a given percentage of fuel removed a lower temperature may be used, provided that the time of heating or the surface exposed be increased. This is an important fact, not only because of possible oxidation of the oil at the high temperature, but also because, after an oil has been heated to 150 or 200 deg. cent., (302 or 392 deg. fahr.), to remove the fuel, it must be cooled before it can be used. The importance of the use of the lowest practicable temperatures for restoring oils is therefore clear.

Conclusions

The possibilities so far shown by the investigation would warrant efforts on the part of automotive engineers toward the development of devices for duplicating the laboratory results in engines, but the data are insufficient to indicate how far the continuous regeneration can be carried in a practical way. A few suggestions as to a possible practical form of this apparatus, which it is hoped to develop in the course of the investigation at the Bureau of Standards, are given in the appendix.

As a conclusion from the data at present available, it can be stated that oils contaminated with different fuels, such as commercial gasoline and kerosene, can be restored entirely or at least their viscosity can be greatly increased by laboratory methods that could probably be incorporated in practical designs for application to present types of engine. Although devices constructed with this in view would give immediate relief, their successful operation might offer increasing difficulties with a further decline in the volatility of commercial engine fuels, which is a very likely condition in view of the steadily rising demand.

For this reason the ultimate solution of the problem lies, in my opinion, along the line of new departures in engine design, preventing the accumulation of liquid fuel in the cylinders and the lubricating system. Although it is difficult to foretell the line of development which will lead eventually to the goal, the combined genius of hundreds of engineers spurred by the need of one of our greatest industries will, undoubtedly, find it. The Bureau of Standards, by undertaking the work in part reported here and by being always in readiness to help solve other problems in cooperation with the industry, is only trying to "do its bit" in the great struggle toward a more efficient utilization of our natural resources.

Appendix

It is obvious that if a device for restoration of lubricating oils which is adaptable to engines now in service, could be put upon the market, a decided saving in fuel cost would result to the users because of the possibility of running on lower-grade gasolines or even on kerosene. For this reason a self-contained appliance that is easy to install would be preferable.

The exhaust gases should be used as the source of heat supply and led through a pipe coil in a small vessel containing the oil or preferably, on acount of ease of cleaning, through a heating jacket surrounding the vessel. The vessel should be connected with the lubricating system and its capacity in relation to the rate of oil feed so chosen that its contents will be completely renewed in 20 to 30 min. A thermostat, of any design now readily obtainable on the market at a low cost could be used to keep the temperature constant at the desired degree. In the case of force-feed lubricating systems, an adjustable by-pass could take care of the oil feed and, by allowing the thermostat to regulate both this and the heating valve, the operation of the device could be rendered practically automatic. A small pipe, connecting the vapor space of the oil container with the intake manifold, would prevent wasting the recovered fuel, and a discharge pipe provided with cooling fins would prevent the return to the crankcase of unduly hot oil. Where there is no oil-pump in the lubricating system, one actuated by manifold suction, such as is now used on some motor cars for the gasoline supply, could be made a part of the appliance.

In newly designed engines a pipe coil heated by the exhaust gases and immersed in the oil-sump, coupled with artificial ventilation of the crankcase, might prove an easy and inexpensive way of retarding undue dilution of the oil. In cases where it is desired to draw air through the oil while it is being heated, the vacuum in the suction manifold could again be usefully employed. This method should be used cautiously, however, as contact with air at high temperatures such as 300 deg. fahr. and over, is likely to cause premature deterioration of the oil.

The saving effected with engines requiring a frequent renewal of the oil on account of excessive dilution, would soon pay for the cost of the additional equipment, even at the present high price of lubricants.

Preignition and Spark-Plugs

By Stanwood W. Sparrow[1] (*Member*)

Annual Meeting Paper *Illustrated with* Drawings

WHAT features of spark-plug construction cause preignition and how does this preignition manifest itself? These two questions this paper proposes to answer. There seems to be no better way of describing the particular phases to be treated than to tell of the events that led us to undertake the work.

After about 4 hr. of what was intended to be a 6-hr. test of a 180-hp. Hispano-Suiza aviation engine, the flame from one of the exhaust ports was noticed to be about twice its usual length and of that light-yellow color which usually indicates a rich mixture. At the same time the power of the engine decreased about 10 per cent, vibration was much increased and the exhaust valve appeared to be extremely hot. The engine was stopped and inspected as thoroughly as possible without dismantling, but except that the spark-plugs were badly carbonized no trouble was found. With new spark-plugs the engine not only completed this run, but made several later runs without any recurrence of this trouble. It should be borne in mind that no adjustments had been made of mixture ratio or throttle during the 4 hr. prior to the appearance of this long yellow flame and that the engine had been developing about 140 hp. during this time. Some weeks later the phenomena again appeared. Temperature measurements were made at this time with a thermocouple inserted in one of the spark-plug holes. This couple showed an average temperature of 150 deg. cent. (392 deg. fahr.) with normal operation of the engine and 400 deg. cent. (752 deg. fahr.) with preignition. The couples used were not constructed for these tests and the actual temperatures have no particular significance as they are temperatures of a part of the cylinder wall thermally insulated from the jacket. It is the relative temperature that is important, namely, that at the time the flame appeared and the power decreased the temperature was much greater than during normal operation. This was confirmed by the fact that when the engine was stopped after a few minutes operation, the exhaust valve in a cylinder that had been preigniting would remain red for nearly a minute after the valves in all the other cylinders had become black. Experiments were then begun to determine the cause of this occurrence and the results of this work are here summarized.

What Is Preignition

Whenever a spark-plug is referred to in this paper as causing preignition, it is the effect of preignition just described that is meant, the long yellow flame, engine vibration, drop in power and extreme heat. The first task is then to show that preignition does produce this effect and to define just what is meant by preignition. Although preignition is quite commonly defined as "self-ignition from the heat of compression," in this case the literal meaning of the word has been more closely adhered to and it has been called ignition from any cause before the proper time, or the time of ignition which will result in maximum power production. It was not difficult to determine that preignition would cause the above noted effects. This was accomplished experimentally by setting one of the two magnetos successively ahead by 15-deg. steps, with the high-tension wire from this magneto disconnected from all but one cylinder. This magneto was not switched in until the engine had been brought up to speed on the other set of plugs. By changing the wires the effect of the advance could be studied in any cylinder. The magnetos are set for an advance of about 20 deg. ordinarily and it was found that firing from about 65 to 215 deg. before center did produce the effects we have described, including the long yellow flame. The intensity of the flame seemed greatest at an advance about midway between these two points and was about the same for all cylinders.

It is evident then that this trouble is caused by preignition. It remains to determine just how the spark-plugs cause preignition. The line of investigation followed was to construct plugs in which different features were exaggerated. The accompanying sketches

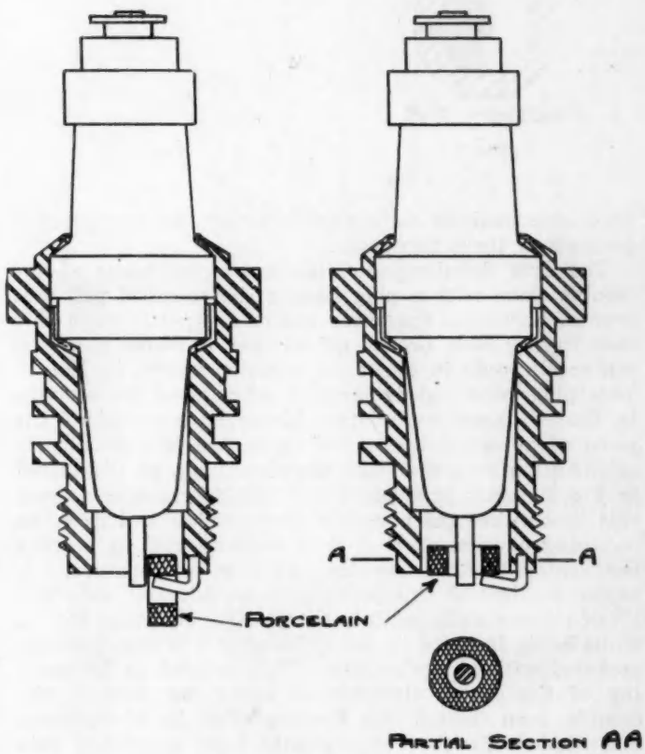


Fig. 1 Fig. 2

illustrate some of these plugs. No special significance attaches to the exact shape of the porcelain or the shell. The plugs in which the special features were incorporated were of two types that have been used with satisfaction in the laboratory and in actual service; and, since both would produce the results noted, the drawings

[1] Associate engineer, Bureau of Standards, Washington.

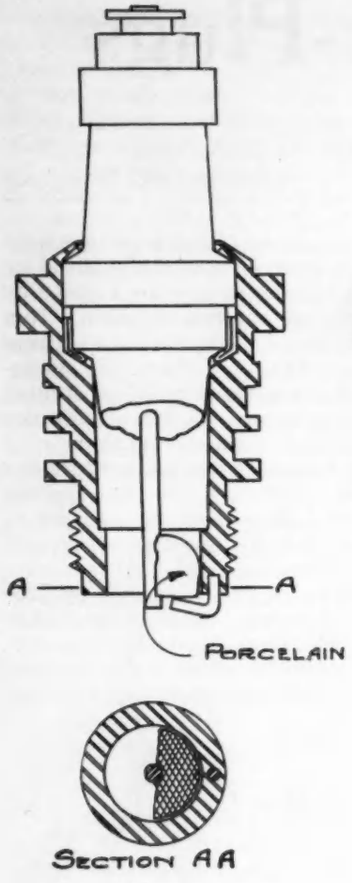


FIG. 3

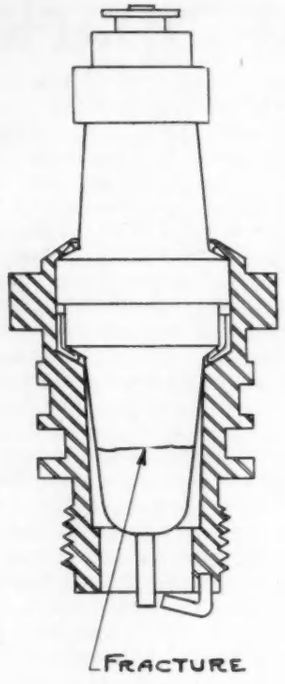


FIG. 4

have been made to show approximately the average proportions of these two plugs.

The first definite indication as to the cause of the trouble came with a plug having the so-called petticoat type of porcelain, the lower end of the petticoat in this case having been broken off so that it rested near the center electrode in about the position shown in Fig. 2. This plug produced preignition when used successively in four different cylinders. Moreover, removal of the piece of porcelain broken off cured the trouble and its substitution on a standard one-piece plug, as illustrated in Fig. 2, again produced it. It would be assumed from this that either the porcelain becomes hot and fires the incoming charge, or that it is instrumental in heating the center electrode to the igniting temperature. A happy accident at this point gave considerable aid. The bit of porcelain slipped into the position shown in Fig. 1, while being inserted in the cylinder and in that position operated without preignition. This pointed to the heating of the center electrode as being the root of the trouble, even though this heating effect be of no great magnitude. Further experiments have supported this supposition, as regards both the cause of the trouble and the narrow range existing between normal operation and preignition. Thus, the plugs shown in Fig. 3, with a piece of porcelain wedged between the center and the side electrodes will persistently cause trouble; while with the piece removed the plug will operate satisfactorily. The breaking away of a large portion of the porcelain from around the center electrode will not change the operation of the plug.

If a plug porcelain be cracked as shown in Fig. 4, so that the break can be detected only by the fact that the center electrode can be moved with the finger, preignition will result. If now the porcelain be broken away, leaving the center electrode bare down to the crack, the plug will again operate without trouble.

Fig. 5 shows a plug upon the center electrode of which has been placed a metal disk of about 5/16-in. diameter and 7/64 in. thick; the spark jumping the annular gap. The plug so constructed caused preignition, but with the disk removed or with one side of the disk flattened as in Fig. 6, it operated normally. Fig. 7 illustrates another case where the line dividing preignition from satisfactory operation is extremely narrow. The plug is unusual only in the respect that the side electrode is replaced by a circular bushing of brass. Several were made leaving the bushing a slip-fit to the shell, all of them causing trouble, while others having the bushing a drive-fit to the shell operated satisfactorily, apparently because of the better heat conduction in the latter case.

What Causes Preignition

Thus far the types have been selected that were of most aid in fixing the cause of the trouble. That cause is an overheated center electrode which preignites the incoming charge and soon causes continuous burning throughout the cycle, extremely high average cylinder-temperature and loss in power. Further examination of the distinctive features of the plugs mentioned thus far shows in each case that these features do cause heating of the center electrode. It would be more direct to say that they prevent the cooling of the center electrode, through preventing its free contact with the cool

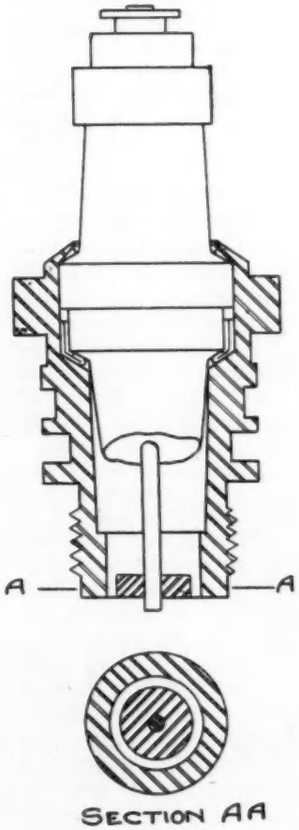


FIG. 5

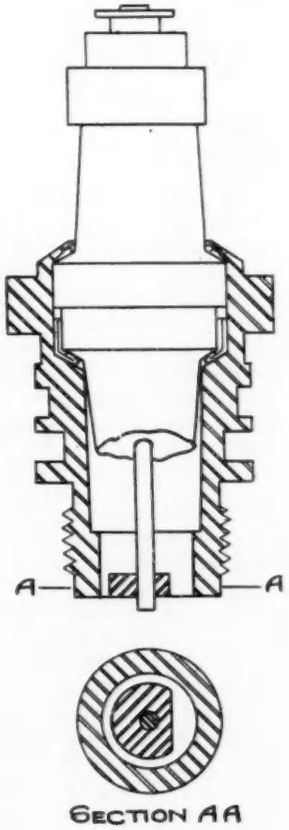


FIG. 6

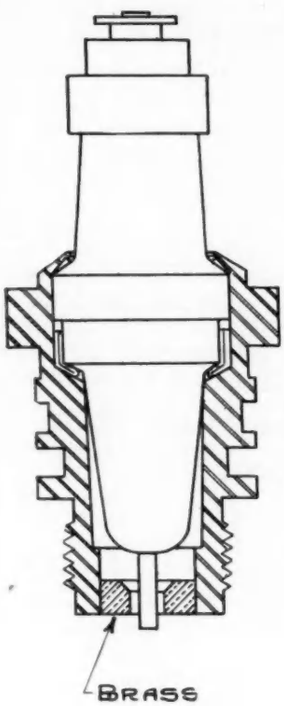


FIG. 7

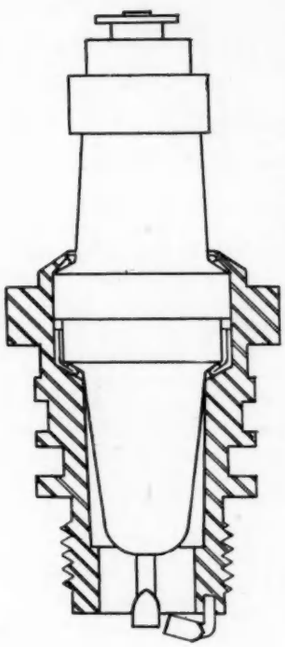


FIG. 8

flashes visible through the window, but with the preignition at the speed of 1800 r.p.m. the observed phenomena were of continuous and rather gradual growth. Thus, with the engine operating normally everything would appear black; then the electrode would appear, become brighter and soon be surrounded by a small spot of reddish flame. As soon as the preignition trouble became pronounced, all of the cylinder that could be observed would appear to be filled with bright yellow flame. The special electrodes shown in Fig. 13 caused preignition and were of sufficient length that they could be easily observed through the window in the manifold. At about 1600 r.p.m. and 80 per cent full load the center electrode would be seen to glow. Under these conditions the engine would operate without trouble, but with a slight increase of load there would be a continuous flame around the electrode and all the evidences of preignition, namely, the long yellow flame, vibration, drop in power and excessive heat would appear. The plug shown in Fig. 14 gave no trouble itself, but when used in a cylinder in which the other plug was one that caused preignition, the nichrome wire, which melts at a temperature of 1500 deg. cent. (2732 deg. fahr.), fused. This is several hundred degrees higher than the average

incoming charge. It is not possible, however, to state whether the trouble produced by the fracture in Fig. 4 is due to the decrease in the heat conducted away by the porcelain or to the free access given to the hot gases by the burning away of the cement around the center electrode.

It is of interest to study a few of the other freak plugs tried in this series. Fig. 8 shows a plug with the electrodes flattened nearly to a knife-edge without producing trouble. Fig. 9 shows an ordinary plug to which has been added a loop of 1/32-in. steel wire extending into the cylinder 2 3/16 in. from the shell. Here again, although there is a wire that doubtless becomes extremely hot, the fact that it is so freely exposed to the incoming cool gases prevents it from attaining an average temperature which is sufficient to cause preignition. Fig. 10 is another illustration of the same fact. In this case a section of a porcelain from a spark-plug is supported on wires extending into the cylinder. Again there is no preignition. Metal disks fitted on electrodes as shown in Figs. 11 and 12 cause no trouble.

The question now arises: How could there be continuous burning in the cylinder throughout the cycle without "popping back" in the carbureter? The explanation is that the rate of flame propagation is less than the velocity of the incoming charge, when the engine operates with the wide-open throttle at a speed of 1800 r.p.m., as in these tests. However, when preignition was present, if the engine was slowed down to 1400 r.p.m. "popping" in the carbureter would appear.

To prove definitely that preignition did produce this continuous burning, a window was constructed in the inlet manifold so that, with the inlet valve open, a portion of the interior of the cylinder could be observed. With a plug such as is shown in Fig. 4, a small section of the side electrode could be seen. When there was "popping" in the carbureter there would be a series of

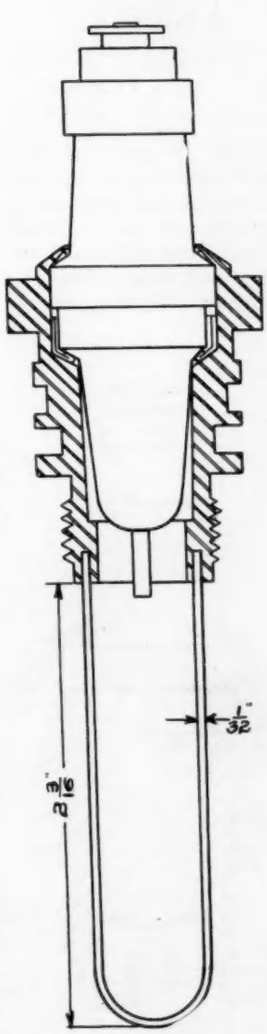


FIG. 9

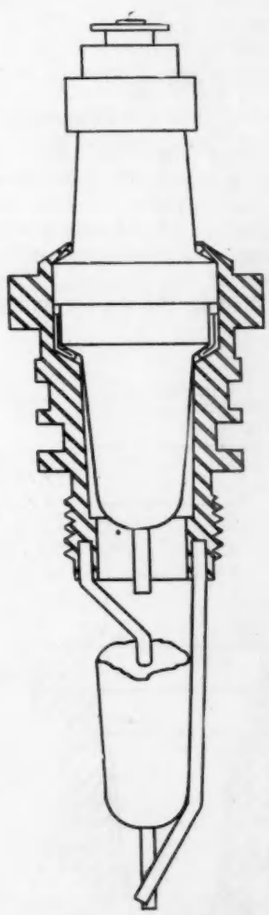


FIG. 10

temperature of a freely exposed wire in a cylinder under normal conditions.

Thus far these remarks have been confined to the "freak" plugs. The question now arises as to why the apparently normal plugs gave trouble. While most plugs that have given trouble have been found to leak around the center electrode, by no means all the plugs that leak here give trouble. Of twelve plugs known to have caused preignition, four were found to leak badly; while of the twelve that operated satisfactorily at the same time, none showed excessive leakage. While it seems reasonable that a leak should cause higher electrode temperatures, the leakage in the above mentioned four plugs seems as likely to have been the effect of the high temperatures, as that it was the cause of the preignition. There is a rather remote possibility that there exists sufficient difference in the heat conductivity of different porcelains to alter the electrode temperature enough to bring about these results. It seems plausible that in many cases the trouble is initiated not by the plug itself but by bits of incandescent carbon or other hot points within the cylinder. The design of the plug determines how long this can happen before the plug attains a sufficiently high temperature to continue the preignition temperature of its own accord. Another point that tends to show this is that some cylinders seem more prolific trouble-makers than others, and that often an adjustment of the mixture ratio away from that of maximum power will prevent the trouble; but once the trouble appears no adjustment will cure it.

Discussion of Results

These then are the outstanding features of this comparatively short series of tests. They serve as a means of identification of this form of preignition and as an indication of the abnormally high temperatures to which valves and combustion-chamber walls are subjected because of this trouble. Finally, they serve as a basis for predicting the performance of a given design of spark-plug or for designing a spark-plug to meet known conditions. Of chief importance in this regard is the fact that, to attain cooling of a spark-plug element, it is more effective to give free access to the cooling action of the incoming charge than to sacrifice this effect to protect the element from the hot explosive gases.

The contradictory demands made upon the spark-plug are realized. Electrodes must be run hot enough to burn clean and prevent fouling at light loads and at the same time be cool enough to prevent preignition at full load. A test similar to this suggests itself as a means for proportioning electrode sizes. It is to design the plug so that each electrode will just produce preignition trouble, at a compression ratio slightly higher than that with which the plug will be used. Care must be taken to avoid badly carbonized cylinders or anything other than the spark-plug that might cause preignition. It appears that in many cases the side electrode can be made smaller and still be less likely to produce preignition than the center electrode, or the latter could be made larger and still maintain a factor of safety against fouling greater than that of the side electrode. That, however, is the problem of makers and users of spark-plugs. It is as a supplement to their own experiments and as a suggestion for their further efforts that these results are presented.

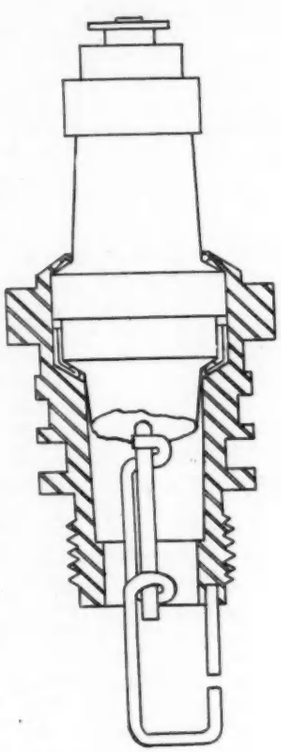


Fig. 13

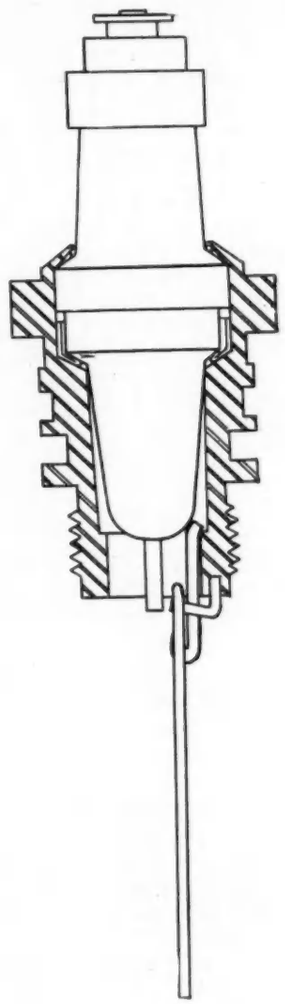


Fig. 14

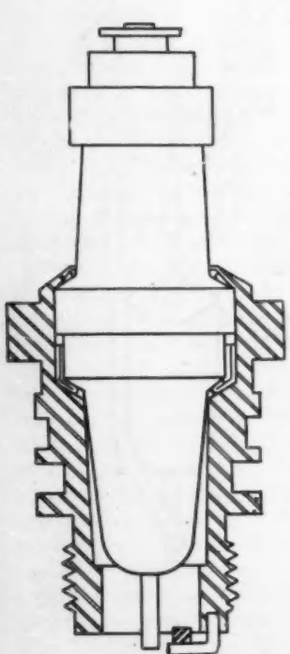


Fig. 11

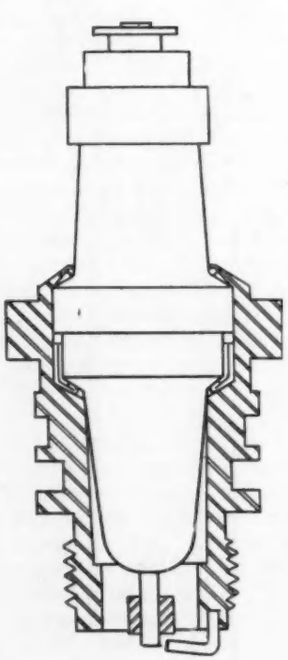


Fig. 12

Tractor Plowing Speeds

By Oliver B. Zimmerman[1] (*Non-Member*)

Tractor Dinner Address *Illustrated with* Charts

AMONG the problems before the designers of plowing tractors, none is of more importance than that of selecting proper plowing speeds. As the variables involved are numerous and elusive, only by careful analysis and experiments can their relative importance be ascertained and the deductions made of value.

The problem is to ascertain the most economical plowing speed at which to operate a tractor to give first-class work at the minimum cost. The solution must be one

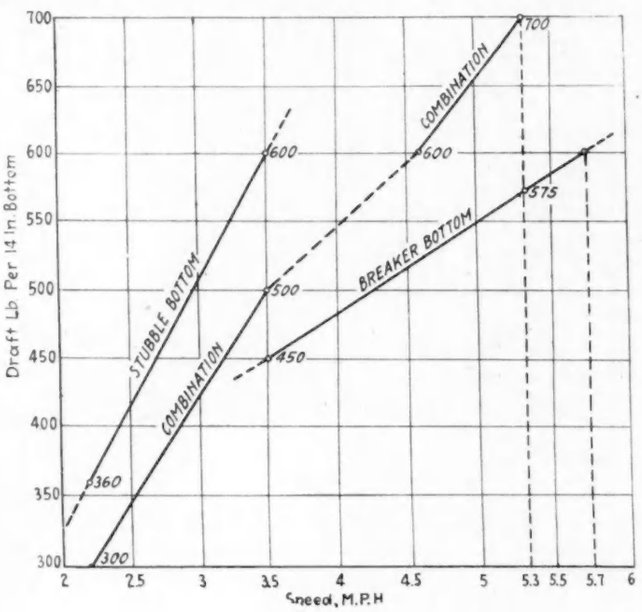


Fig. 1

which will be right from the manufacturer's standpoint as well as from that of the farmer. This means that a variety of soil resistances, different speeds, widths and depths of cut, forms or shapes of plows, etc., must be considered. It will be necessary to devote most of this paper to the question of what becomes of the power delivered at the drawbar and to assume certain standard conditions about which the argument can be confined.

First, taking a high-angle view of the situation to simplify the analysis and using two well-designed outfits, one developed to travel at 2 m.p.h. and draw four plow bottoms, and the other drawing two plow bottoms at 4 m.p.h., if it is assumed for the moment that the drawbar horsepower is equal and that the acreage accomplishment in the same time is equal, what conclusions can be drawn from the comparison? It is evident that the tractor must move itself and the plows over the ground twice as far in one case as it does in the other, that the number of turns at the ends in one case is double that of the other and that the strains due to striking hidden obstructions are greater at the higher speed. These are the factors which, in the main, require more horsepower ahead of the drawbar at the higher than at the lower speeds, and proper care in design.

Next consider what occurs at the rear of the drawbar. Here it is assumed to cover a reasonable variety of soil resistance and number of plows, that 15 drawbar hp. is available. It is also assumed that for each speed from 1 to 5 or 6 m.p.h. the engine and tractor are properly designed for the speed considered and of equal fuel economy per horsepower above and below each mile of speed, so that the merging shall be complete and uniform from one into the other and the comparison as fair as possible.

Increased Speed and Draft

For basic data the recently published draft data of Prof. Davidson of Ames, Iowa, and those of the Kansas State Agricultural College are used, as these tests are in complete agreement with experimental data developed by experienced commercial organizations in the past and used by them in present-day designs. The data mentioned indicate in general that in each kind of soil, whether heavy or light, with the increase of speed there is a corresponding increase of draft, the amount of which is dependent upon the speed, shape of plow and nature of soil.

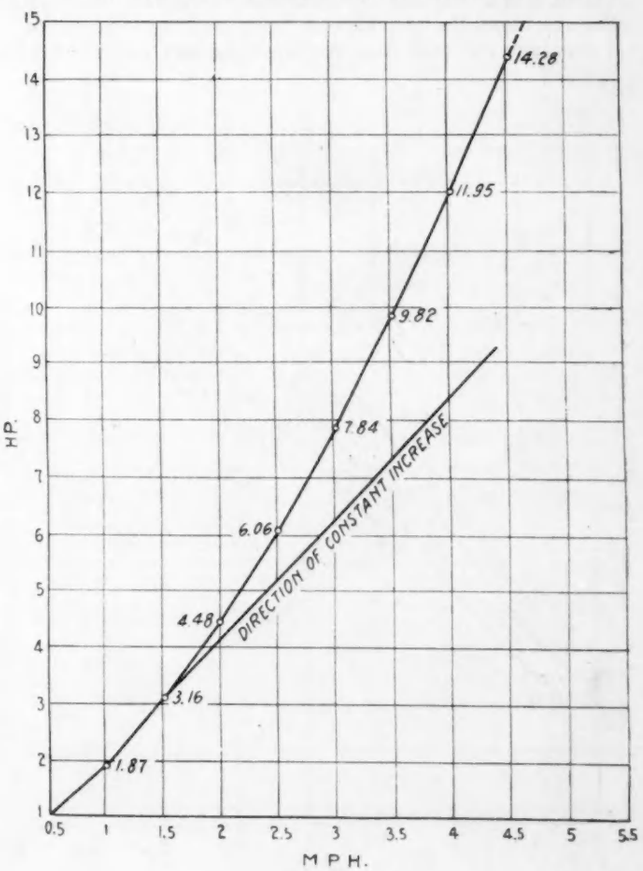


Fig. 2

[1]International Harvester Co., Chicago, Ill.

Fig. 1 is typical and represents much of interest in that between the limits of the experiment, over a range of speeds not too far from the designed rates for the various kinds of plowshare, the plows show a uniform increase in draft with an increase of speed and the draft is illustrated by a straight line. The combination plow bottom was experimented with at two speed ranges. It should be borne in mind that the soil resistance is very small in these Kansas tests. In percentages, with a stubble plow bottom in light soil, for an increase of 1.3 m.p.h. above 2.2 m.p.h., the draft increase is 240 lb. or, stated in percentage, a 59 per cent increase of speed causes a 75 per cent increase in draft. For the combination plow bottom there is a similar set of figures; a 59 per cent increase of speed causes a 66 per cent increase of draft. With the breaker plow bottom, a 16.3 per cent increase of speed, 3.5 to 5.7 m.p.h., causes a 33 per cent increase of draft, 450 to 600 lb. The deduction is clear. With the increase of speed there is a proportional increase of draft and the less abrupt moldboard has the least resistance. The same is shown in Prof. Davidson's data (See Fig. 2), where an increase from 2 to 4 m.p.h., or 100 per cent, causes a draft increase of 33 per cent.

Next taking up the horsepower required to perform the operation and translating the speed and draft into power, it will be noted from Fig. 2 that not only must the work due to the added speed be accomplished but in addition the added percentage of draft must be provided for. Hence, it is found on this chart that in speeding up from 2 to 4 m.p.h., a 100 per cent increase of speed requires an increase from 4.48 to 11.95 hp. per plow, or 267 per cent.

Fig. 3 is a diagram of horsepower required, illustrating the relations for the Kansas tests and their position to a constant increase line, emphasizing again that at both low and high speed the effect of increased speed is to reflect itself in much increased horsepower.

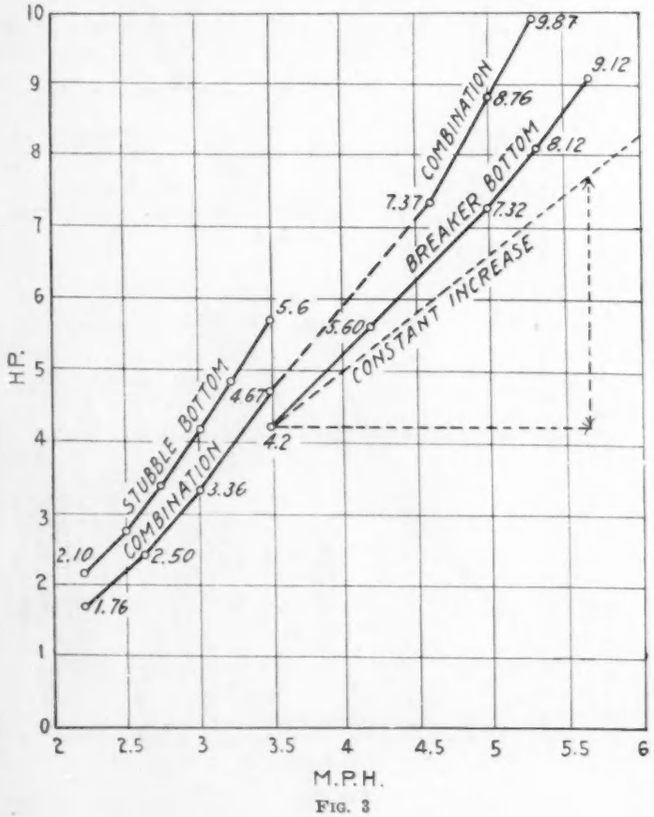


Fig. 3

Fig. 4 was developed from the tests, giving a comparison of various sizes of tractor operating at various speeds and in various resistances of soil. Data were obtained by determining at the various speeds how many plows the tractor indicated would pull. By cross-reading from any given soil resistance at the left and from a point indicating the number of plows, the speed at which they

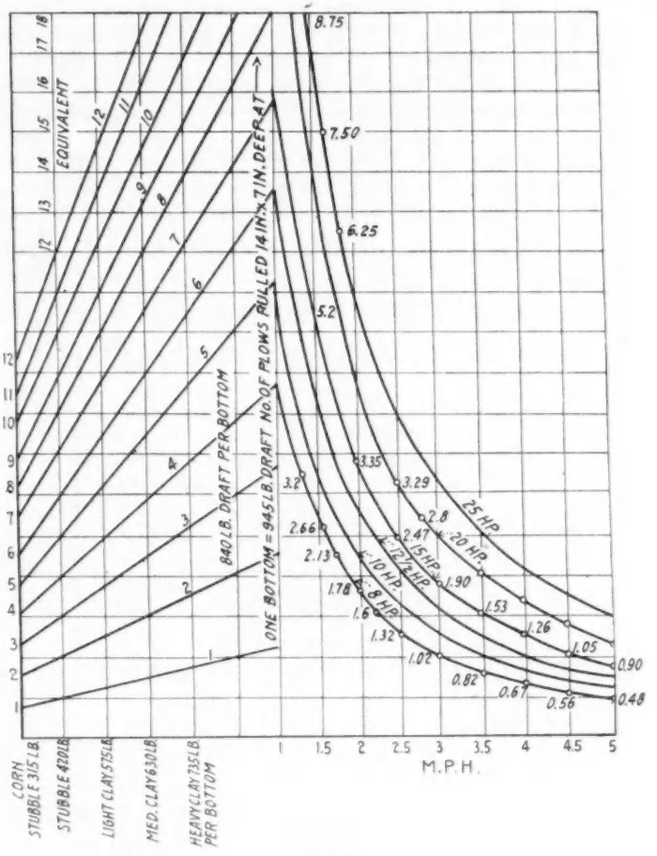


Fig. 4

can be pulled is shown at the curve; 575-lb. soil at 2 m.p.h., four plows, 15 drawbar hp. curve, reads 2½ m.p.h. Likewise, a given tractor speed followed to the curve and then to the left to intersect a vertical line, records the full number of plows to pull in that soil; 2 m.p.h., 10 drawbar hp. curve, 630-lb. soil at 2 m.p.h. reads three plows.

Area Plowed at Different Speeds

With the data indicated based upon careful tests in light soil, the acreage plowed per hour by the tractor at various speeds and in various soils can be plotted as in Fig. 5. By calculation, the curve swinging downward and to the right represents the curve of maximum area or acreage which can be plowed economically at the speeds shown, in the soil indicated and with the designated 15 drawbar hp. Intersecting this curve are various radial lines designating the path on the field of various numbers of plows cutting 14 in. in width. A different series of radial lines would be drawn should 10 or 12 in. or other bottoms be under consideration. This chart is very instructive because it explains many factors which are confusing or hard to explain when field accomplishments are analyzed.

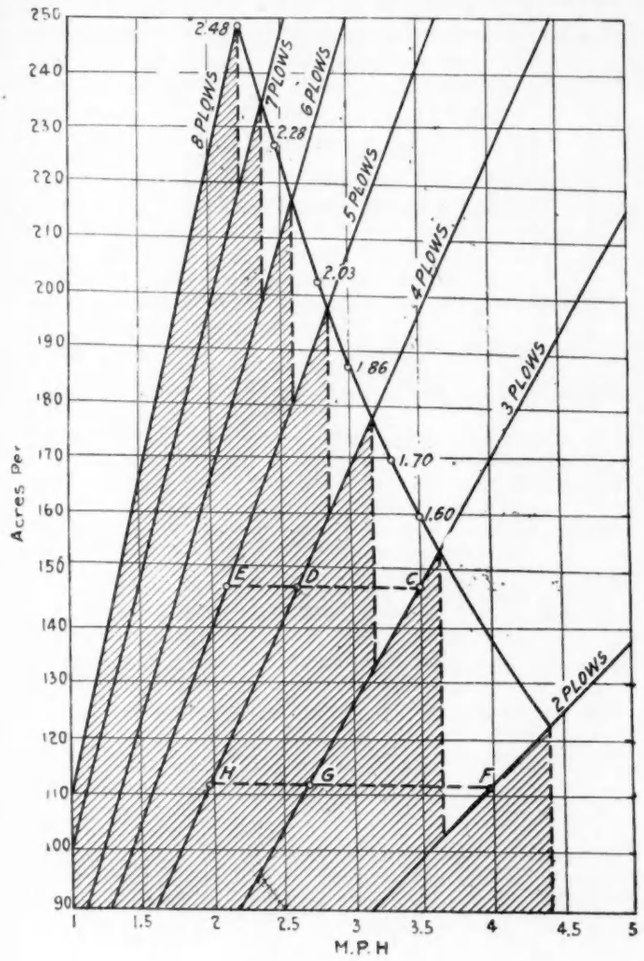


FIG. 5

The sawtooth lines indicate that a 15 drawbar hp. engine, in this soil, starting at 2 m.p.h., would begin by plowing at the rate of 2.20 acres per hr. The acreage would increase with an increase of speed up to 2.50 acres per hr., when a speed of about 2.25 m.p.h. has been reached. At this point the drawbar horsepower has reached 15 and a plow is dropped off, continuing with seven plows, from 2.20 acres. Again, as the speed advances, the acreage increases until the seven-plow line cuts the maximum when another plow is dropped, and so on.

Now considering that three tractors are at work under the same conditions and drawbar horsepower, operating three, four and five plows, which are designated respectively as *C*, *D* and *E*, at such speeds as will cause each to plow 1.47 acres per hr., what can be deduced from the location of these three points? First, the three-plow rig is working nearer its maximum than the four or five-plow rigs, since the distance from the maximum is increasing or, in other words, *C*, *D* and *E* are not requiring 15 drawbar hp. and the five-plow outfit is using the least horsepower of the three to plow the same acreage in the same time. Utilizing the full 15 drawbar hp., the acreage is in the relation of three plows to 1.54 acres, four plows to 1.78 acres and five plows to 1.97 acres.

Again, taking the point *F*, a two-plow outfit operating at 4 m.p.h. covers 1.13 acres while the same outfit fitted with eight plows would cover at 2 m.p.h. 2.22 acres or nearly 100 per cent more acreage. The maximum condition using 15 drawbar hp. with two plows is 1.25 acres, while with eight plows it is 2.48 acres, approximately 100 per cent more acreage.

Admitting that this diagram does not compensate for the increased weight drawn of the eight over the two-plow rig, nor for the fact that the plow shape is not correct for the widely different speeds, nevertheless, it is positive that as the comparative speeds decrease so that the two are only small percentages above and below the designed speed for the plow under given conditions, that the same relation holds true. The corrections mentioned would tend to reduce somewhat the acreage at the lower speeds to an extent covered by the power absorbed in hauling the added weight of the greater number of plows over the ground, and the greater speeds would reduce the available plowing energy.

Where heavy land is under consideration, much the same conditions are met and analyzed in Fig. 6. It is clear from both Figs. 5 and 6 that the *acreage plowed decreases markedly with increased speed* to such an extent that correction in moldboard design or reductions in weight cannot overcome the deduction *that the lower speeds are more desirable than the higher.* It is also clear that time is not gained by high speeds; in fact, time is lost.

Occasional analyses of field demonstrations have led to wrong conclusions because the machines were not being compared on an equal basis, a condition quite impossible to attain except theoretically as shown. Two cases to

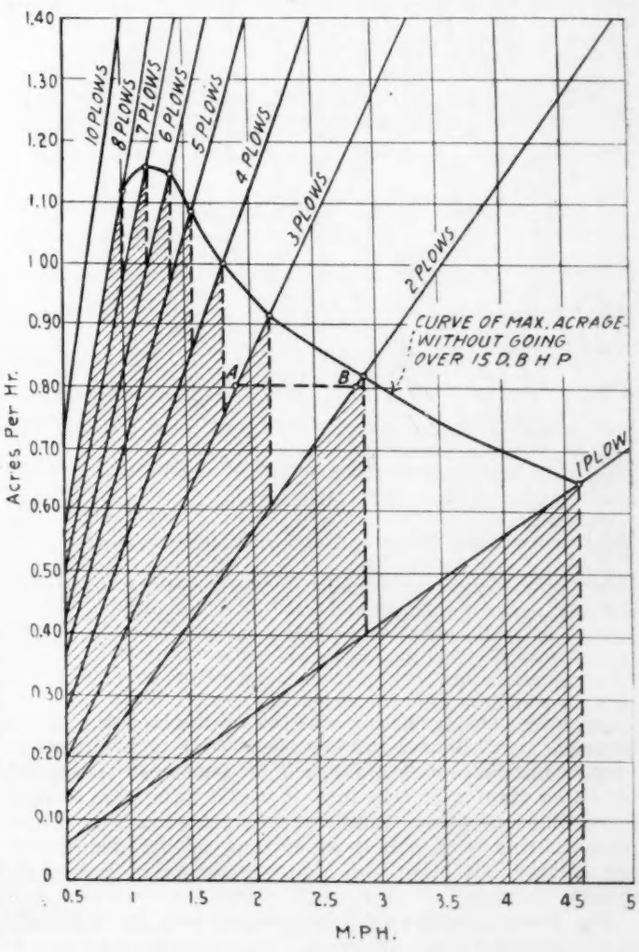


FIG. 6

illustrate this in heavy land are detailed in the accompanying table.

Tractor	Number of Plows Pulled	Gear	Speed, m.p.h.	Area Covered, acres
1	3	Low	2.20	0.92
2	2	Plowing	2.75	0.80
1	2	Plowing	2.75	0.80
2	3	Low	1.75	0.75

This indicates that the ability to win, in a plowing contest, for instance, is dependent upon the relations which exist at the time and not necessarily upon the efficiency of the engine and outfit, but upon the relations of the possibilities of adjusting the speed and number

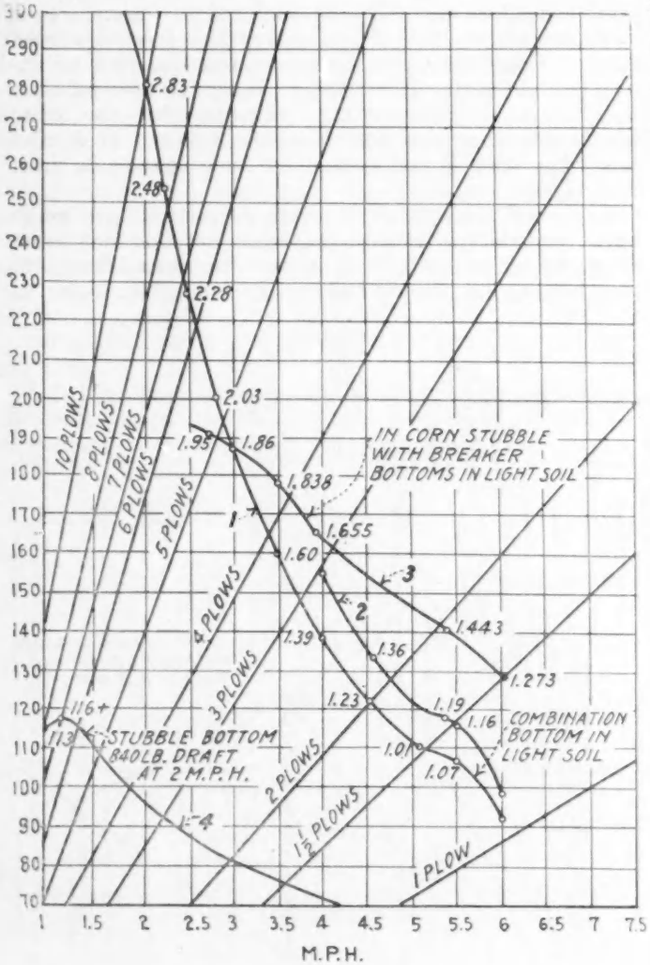


FIG. 7

of plows so as to operate close to the line of maximum acreage. In this case the low speed wins once and loses once, but if the speed of 3 m.p.h. or over were attempted by either tractor in this land it would be unable to pull more than one plow. Similar comparisons in Fig. 5 are more in line with actual experience, as the deductions are clearer, although this soil resistance is less than usually obtains.

Fig. 7 combines the various acreages with the different plow bottoms in one diagram. Aside from the correct determination of the proper plowing speed, there arises

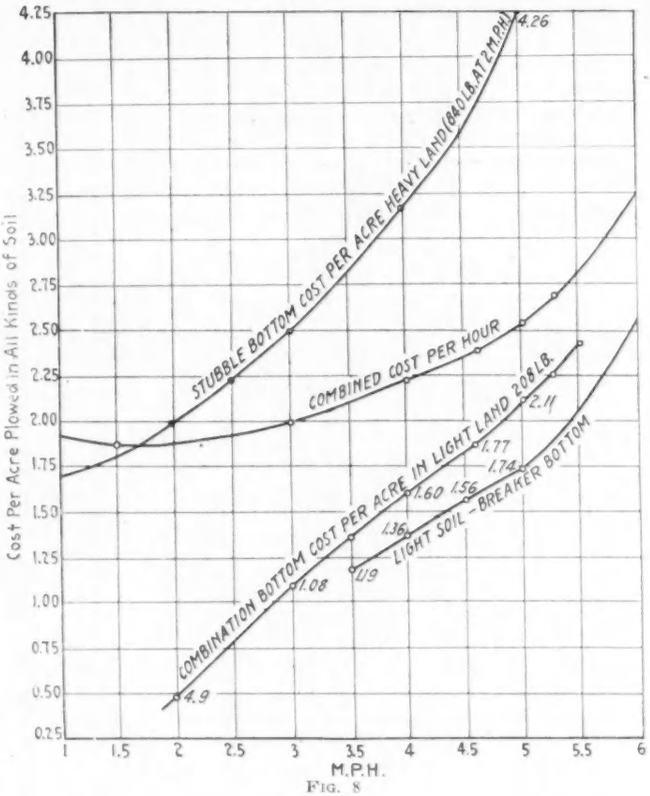


FIG. 8

the question of the desirable speed for hauling equipment such as threshers and wagons over the road, the drawing of mowers, binders and similar farm equipment in actual operation. This question is outside of the discussion. New apparatus could and will be designed for higher

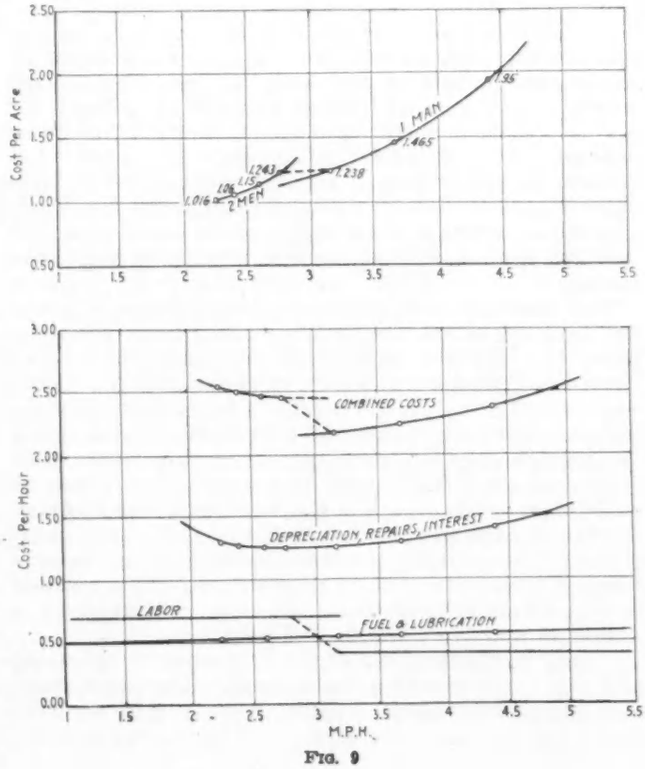


FIG. 9

speeds, but it must be evident to those with experience that the percentage of increase in economic workable speeds cannot be far from those deduced from this analysis.

Cost of Operation

Data on this subject have been used in plotting Fig. 8, which are indicated by experience covering the following:

(1) First cost of 25-hp. tractor, plows, tanks and supplies, a constant, $2,500
(2) Interest on the investment at 5 per cent
(3) Depreciation on the basis of 8½ years at 2 m.p.h. to 6 years at 4 m.p.h.
(4) Repairs, on an increasing rate from 6 per cent at 2 m.p.h. to 8½ per cent at 4 m.p.h.
(5) Labor, at 40 cents per hr. and 30 cents per helper on over four plows
(6) Fuel and lubricant for quantity on the basis of 5 per cent added for each mile of increased speed, at the market prices and on the basis of 65 per cent transmission efficiency at the drawbar
(7) The hours of operation to be figured at 500 per year, based upon U. S. Government records

It is clear that the cost rises rapidly with the speed. The same facts have been used in Fig. 9 introducing the two-man operator feature and charging the actual cost of plows, engine and equipment used, as compared to the constant cost shown in Fig. 8.

A complete development of draft curve, horsepower curve and number of plow curve with relation to miles per hour, is here combined in one chart in Fig. 10. From this, therefore, the conclusions follow that

(1) Placed upon a comparable basis and with reasonable assumptions supplementing the carefully prepared and conducted experiments used, the most economical plowing speeds are unquestionably below 3 m.p.h., the cost rising rapidly from about 2 m.p.h.
(2) None of the variables which have been left out of this discussion can enter into the question to a degree that will overcome the evidence indicated by these charts
(3) If plows were designed to meet higher speeds they can help only in reducing the greatly increased costs at the higher speeds, but they cannot by any means veer the economical point above 3 m.p.h.
(4) The argument for better breaking up of the soil at the higher speeds is heavily paid for, and it is more economical to perform the operation of harrowing either separately or behind the plows than to try to accomplish it by rapid plowing
(5) To attain the very best results with the plowing outfits, speed ratios of plow gearing and throttle should cover the ranges from 1¾ to 3 m.p.h. This would permit the most flexible ranges to meet the various soil resistances (See Fig. 4)
(6) From Fig. 6 it appears that in heavy land and in plowing up to 8 in. in depth with small tractors, greater acreage would be attained by the use of 10 or 12-in. instead of 14-in. plow bottoms, since more of the sawtooth points will thereby meet the economical acreage curve. A better adjustment of cutting width is also possible
(7) The time required to plow 1 acre can be tabulated

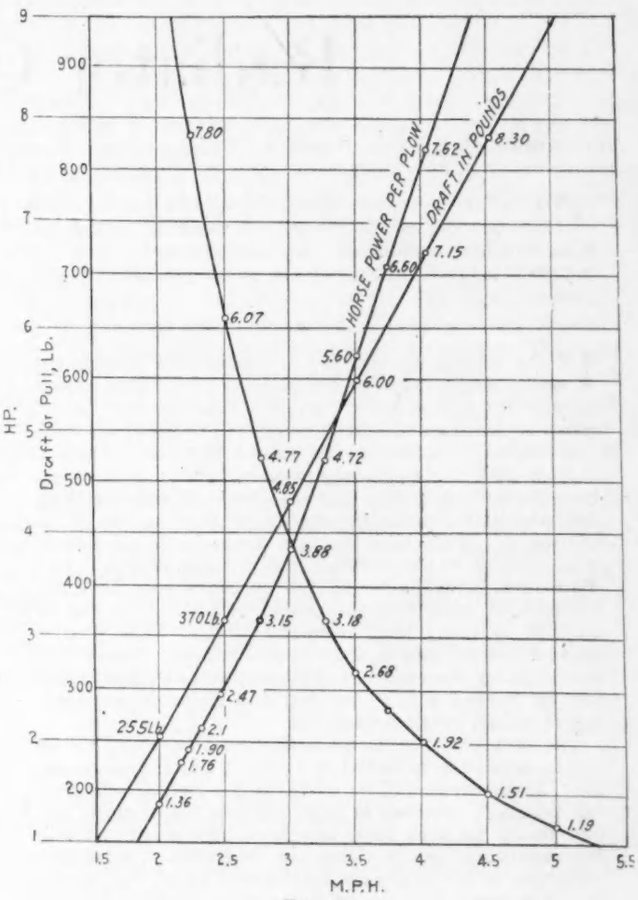


Fig. 10

from these tests, with relation to the number of plows pulled, as follows:

Number of Plows	Heavy Land, hr.	Light Land, hr.
2	1.212	0.643
3	1.095	0.565
4	1.000	0.506
5	0.926	0.462
6	.....	0.426
7	.....	0.400

(8) The low limit of plowing speed is obviously the speed which will permit the proper scouring of the moldboard. This speed will be dependent upon soil conditions. The proper moldboard is assumed to have been chosen, i. e., one whose abruptness and scouring qualities are correct for the conditions to be met
(9) The cost diagram, using two operators at low speeds and with five or more plow bottoms, can be further cut by using power lift equipment where one man can operate up to eight plow bottoms
(10) This presentation indicates that much needed work can be carried out experimentally to improve both engine and plows, with results of considerable economic value to the industry, and it should cause a series of important discussions to follow the analysis.

Radiator Cooling Fans

By GEORGE W. HOYT[1] (*Member*) MINNEAPOLIS SECTION PAPER

THE following written discussion of this paper, which was printed in the May, 1919, issue of THE JOURNAL, has been submitted. For the convenience of the members a brief abstract of the paper precedes the discussion.

ABSTRACT

IN the cooling system for an automobile engine, the water-jacket must be designed to give ample capacity and free flow of the water. It is essential that water-pump capacities and speeds be figured to equal the radiator capacity so as not to retard the flow of water through the radiator and cause hot water to back up into the cylinder; the radiator must always be kept full and still handle the water as fast as the pump carries it. Fan locations are necessarily considered with relation to the radiator and radiator shroud. Fan diameters, blade path and fan speeds should be given thought, in order that the proper volume of air can be handled to carry heat from the engine. The frontal area of the radiator core in square feet per horsepower developed by the engine and several other details which can be worked out by the fan and radiator manufacturers should receive attention.

The best possible fan bearings must be used, giving special attention to radial and thrust loads, fan speeds, etc. Attention should be given to the method of oiling fan bearings, whether of ball, plain or roller type; the hub should be kept tight and free from dust. Strong fan mountings or brackets are desirable to minimize vibration which is always detrimental to all fan parts. In other words, instead of using what space is left on the chassis for fan purposes, far better results will be gained if the foregoing items are given proper attention, thus dissipating the heat produced by the engine in a satisfactory manner. All necessary information on this subject is at present available through the co-operation of car and accessory manufacturers.

THE DISCUSSION

LOUIS SCHWITZER:—Mr. Hoyt's statements with regard to end thrust appear questionable to me and the charts showing the amount of it seem somewhat misleading. He says that a fan running in free air will produce a certain end thrust, but that when this fan is placed back of a radiator the end thrust is increased about 25 per cent, depending upon the restriction of the air passage. It is a fact established upon a physical law that the pressure exerted by a moving air column, the end thrust, is a direct function of the air velocity. In other words, with an increase of the air velocity the pressure, or end thrust, increases. If a fan is placed behind a radiator, the air velocity produced by the fan is considerably less than in free air, on account of the air-passage restriction. If Mr. Hoyt's contentions are correct, with the air intake restricted to such an extent that no air could be delivered, the end-thrust load would be infinitely large. This would be unreasonable, because the pressure exerted by a moving air column per square foot is

$Pt = 5.2\ h$; where
P = the pressure in pounds
t = the time in seconds
h = the velocity-head in inches of water

[1]Chief engineer, Oakes Co., Indianapolis, Ind.

The velocity of air in feet per minute is

$$V = 4000\sqrt{h} \text{ or, } h = \frac{V^2}{16{,}000{,}000}$$

From this formula

$$Pt = 5.2 \times \frac{V^2}{16{,}000{,}000}$$

This shows that the pressure exerted by a moving column of air is practically in direct proportion to the square of the velocity.

Mr. Hoyt states that he made the tests to determine the end-thrust load by using an indicator on the end of a 5-hp. electric motor shaft, to which the fan blade was directly coupled. In my opinion the only way to measure the end-thrust load on a propeller-type fan is to measure it directly on the fan. This can be done either by measuring the pressure of the air column delivered by the fan, or by mounting the fan movably on its spindle so that it exerts its pressure on a scale, through its motion in the opposite direction from that of the air current. It is also a very easy matter to determine the end-thrust loads by the horsepower input to the fan and the air velocity thus produced. The charts in the original paper also show that the end thrust drops faster than does the air velocity; in other words, by restricting the inlet the end-thrust load will be diminished faster than the air velocity diminishes. All of these curves show end-thrust loads of fans moving in free air. The manufacturer is more interested in knowing what end-thrust loads are produced in a fan behind a radiator, than in those obtained in free air or in a wind tunnel.

From actual tests and measurements on nearly 400 different makes of passenger cars, trucks and tractors, I have found that on about 80 per cent of these the maximum average air velocities obtained through the radiator are below 1500 ft. per min., on about 17 per cent they are between 1500 and 1900 ft. per min. and on only 3 per cent are they above 2000 ft. per min. The highest velocity I ever found was 2200 ft. per min. These are the air velocities that must be taken into consideration in determining the end-thrust loads in radiator fans under actual operating conditions. It will be found from Mr. Hoyt's charts that on a 16-in. fan, considering even the maximum obtainable average air velocity of 2000 ft. per min., the end-thrust load will not be more than 3 lb.; on an 18-in. fan it will be about 3¼ lb., and for 20 and 22-in. fans Mr. Hoyt does not show air velocities as low as the maximum velocities actually obtained with these sizes of fans on automotive vehicles. In the case of the 22-in. fan the velocity and end-thrust curves start with 2600 ft. per min. and a corresponding end thrust of about 6.8 lb. I believe that no truck or tractor can be found where air velocities of 2600 ft. per min. through the radiator can be measured. In passenger cars, where this or higher air velocities through the radiator can be produced, it is only done at high vehicle speed. At these speeds, however, the initial air velocities produced by the motion of the vehicle tend to drive the fan and balance the end thrust produced by the fan itself. In any event, it is

shown by the charts that the end-thrust loads in radiator fans for the air velocities actually produced in automotive apparatus, will not exceed 2 to 3 lb. in 16-in., 3 to 4 lb. in 18-in. and 4 to 5 lb. in 20 and 22-in. fans. This is one of the reasons why plain end-thrust arrangements have stood up so remarkably well in the field, as well as in laboratory tests up to an equivalent of 100,000 miles of driving.

With regard to the bearing question, a great many of the cup-and-cone type have been used, but their life is short, especially when they are abused. To drop a fan with cup-and-cone bearings upon the floor, or to drive it with only slight force into its supporting bracket, means the destruction of the races. Such bearings are also adjustable and, therefore, the matter of adjustability enters into the question. It takes an expert to know when these bearings are properly adjusted, and when not in proper relation with each other they are noisy and deteriorate rapidly. Slight wear lessens the efficiency of such bearings and carrying an end-thrust load in addition to the radial load results in considerably less radial-load carrying-capacity, which is the main load to be taken care of in cooling fans. The same arguments apply to annular-ball end-thrust bearings. Plain end-thrust bearings with an automatic adjustment according to the end-thrust load are the most successful according to a test recently made in which the plain end-thrust bearing ran for 109 hr. at 1500 r.p.m., without lubricant, with only a few score-marks, while the cup-and-cone bearing lasted only 11 hr. under exactly the same conditions.

Mr. Hoyt states correctly that about one out of ten people who drive cars ever oils the fan. Much less will they adjust stuffing-boxes, especially when these require special-size wrenches and are inaccessible. With adjustable stuffing-boxes the felt invariably squeezes out between the shaft and the stuffing-nut, thus providing a wick oiler to drain the oil to the outside. Self-contained felt arrangements on the inside of the hub, which throw the oil they wipe off the shaft back into the bearing are, in my opinion, the most simple and successful type.

With regard to the distance of the fan from the radiator, it is not practicable to define this distance specifically. The most efficient distance depends upon the type of radiator core with regard to its air resistance, upon its size relative to the diameter of the fan and whether the fan is with or without a shroud. In the tests at Washington, with which I was connected, I found, for instance, that a 20-in. fan behind a 25 by 26-in. radiator showed highest air velocity through the radiator at a distance of 1½ in., while the same fan, within a shroud, behind the same radiator, showed the maximum efficiency at a distance of about 3 in. from the radiator. This efficiency was barely influenced by moving the fan either 1 in. toward or away from the radiator core. I contend that the best distance should be determined by experiment in each individual case.

KANSAS CITY TRACTOR DINNER

THE fourth annual Tractor Dinner of the Society held at Kansas City on Feb. 19, during the week of the National Tractor Show, was an instructive as well as interesting and entertaining event. Two hundred and fifty members and guests were present, all imbued with the spirit of the impressive exhibition staged by the Kansas City Tractor Club in the form of the show and of the vitally important design, production and economic problems now before the tractor engineers for solution.

E. E. Peake, of Kansas City, presided as toastmaster in a very amusing and satisfying way.

President J. W. Perry, of the Kansas City National Bank of Commerce, addressed the members in a stirring manner and held their interest intently. He told of his own personal experience in the operation of farm tractors and of the confidence that the banks have in the tractor business. He stated that the tractor sales in Iowa, Nebraska, Missouri, Oklahoma and Kansas amounted to $90,000,000 last year, citing the further fact that there are approximately 91,000,000 acres of agricultural land in cultivation in this territory. He believed that of the 125,000 or more tractors in use, 50 per cent were in the region mentioned. Mr. Perry predicted that high prices in general would cease.

Guy Hall, manager of the National Tractor Show, made some felicitous remarks, stating generously that the continued and increasing success of the annual Tractor Exhibition was due in large part to the cooperation of the Society members.

M. B. Morgan, formerly a major in the service of the Ordnance Department, gave the members a view of some films selected by the Ordnance Department as representative of the work of its automotive equipment.

R. F. Crawford, president of the Kansas City Tractor Club, talked of the needs of the tractor industry as seen by the sales department. He entered a strong plea for the encouragement of development resulting in a few conventional types of tractor, as in the case of automobile production. He also pointed to the automobile industry as a good example of broad cooperation.

O. B. Zimmerman, formerly a major in the Engineer Corps, explained briefly some studies recently made by him. He illustrated vital facts in the matter of the best working speeds of tractors. He based what he had to say upon the principle that a great industry like that of the tractor should be founded on careful investigational and research work. Mr. Zimmerman gave some very important deductions from a coordinated consideration of the various data recently published in connection with experiments on plow draft. His address and the curves referred to therein are included elsewhere in this issue of THE JOURNAL. It is clear that it deserves and there is little doubt that it will receive careful attention from tractor designers.

Applicants *for* Membership

The applications for membership received between Jan. 24 and Feb. 28, 1920, are given below. The members of the Society are urged to send any pertinent information with regard to those listed which the Council should have for consideration prior to their election. It is requested that such communications from members be sent promptly.

AMBROSE, FREDERICK S., chief engineer, Singer Motor Co., Inc., North Third Avenue, *Mount Vernon, N. Y.*

BARLOW, ROLAND WILCOX, production manager and consulting engineer, Hamlin-Holmes Motor Co., 29 South La Salle Street, *Chicago, Ill.*

BASSOFF, ABRAHAM, designer, Engineering Service Corporation, *Detroit, Mich.*

BENTLEY, G. P., layout draftsman, Supreme Motors Corporation, *Warren, Ohio.*

BIRCKELBAW, LLOYD I., mechanical engineer, San Joaquin Light & Power Corporation, *Fresno, Cal.*

BIXBY, GEORGE L., research engineer and service manager, Detroit Electric Car Co., *Detroit, Mich.*

BLAINE, W. E., automotive engineer, Cleveland Tractor Co., *Cleveland, Ohio.*

BLAKEMORE, THOMAS L., aeronautical engineer, Goodyear Tire & Rubber Co., *Akron, Ohio.*

BOCKMANN, ARTHUR C., automobile electrician, Union Automobile Co., 130 Columbia Street, *Union Hill, N. J.*

BOOCK, ALFRED C., draftsman, Pan Motor Co., *St. Cloud, Minn.*

BOOTH, EARL C., chief engineer, Holcomb & Hoke Mfg. Co., *Indianapolis, Ind.*

BREESE, ROBERT P., assistant engineer, Loening Aeronautical Engineering Corporation, *New York City.*

BURNS, WILLIAM T., engineer, Timken Roller Bearing Co., 1309 Ray Place, Northwest, *Canton, Ohio.*

CASLER, HERMAN, vice-president and consulting engineer, Watson Products Corporation, *Canastota, N. Y.*

CLUCAS, W. J., sales representative, Lancaster Steel Products Corporation, *Lancaster, Pa.*

COULTER, JAMES, vice-president, Automatic Machine Co., *Bridgeport, Conn.*

CRAWFORD, MELVIN W., sales engineer, Edward A. Cassidy Co., 280 Madison Avenue, *New York City.*

CRUTCHFIELD, WILLIAM H., aeronautical mechanical engineer, engineering division, Air Service, McCook Field, *Dayton, Ohio.*

DEARBORN, RICHARD JEWELL, patent attorney, Texas Co., 17 Battery Place, *New York City.*

DOUGHERTY, E. FRED, draftsman, Rome Wire Co., Rome, N. Y., and E. D. & A. F. Cronk, *Utica, N. Y.*

DOW, HAROLD T., superintendent, Spring Perch Co., *Stratford, Conn.*

FAIRBANK, L. G., vice-president and manager, Firestone Steel Products Co., *Akron, Ohio.*

FISCHER, CHARLES L., master mechanic, Willard Storage Battery Co., East 131st Street, *Cleveland, Ohio.*

GAETH, PAUL, president and general manager, Gaeth Motor Co., 2101 Abbey Avenue, *Cleveland, Ohio.*

GOLDSMITH, HENRY, branch manager, G & O Mfg. Co., *New Haven, Conn.*

GOODRICH, R. H., chief engineer, Powrlok Co., 1107 East 152nd Street, *Cleveland, Ohio.*

GORING, RICHARD G., engineering representative, Fellows Gear Shaper Co., *Springfield, Vt.*

HALL, DONALD A., aeronautical draftsman, checker and layout man, Curtiss Engineering Corporation, *Garden City, N. Y.*

HARRISON, EDWARD F., owner and manager, Harrison Motor Car Co., 1726 Glenarm Street, *Denver, Col.*

HARROD, VAUGHN C., draftsman, Kelly-Springfield Motor Truck Co., *Springfield, Ohio.*

HASKINS, L. O., vice-president and sales manager, Powrlok Co., 1107 East 152nd Street, *Cleveland, Ohio.*

HILL, L. CLAYTON, consulting engineer, 314 Holden Building, *Detroit, Mich.*

HILL, LYMAN L., aeronautical designer, McCook Field, *Dayton, Ohio.*

JACKSON, EDWARD WILLIAM, automotive engineer, Post and Telegraph Department, *Wellington, N. Z.*

JACOBSON, C. A., JR., designing engineer, Jacobson Engineering Co., Inc., 30 Church Street, *New York City.*

JONES, G. D., agricultural engineer, Cleveland Tractor Co., *Cleveland, Ohio.*

KAPLAN, MICHAEL J., aeronautical draftsman, Naval Aircraft Factory, League Island Navy Yard, *Philadelphia, Pa.*

KARGAU, THEODORE, production engineer, Batholomew Co., *Peoria, Ill.*

KEESLER, CLYDE C., secretary and assistant general manager, Watson Products Corporation, *Canastota, N. Y.*

KELLEY, EDWARD J., chief draftsman, Abbot-Downing Truck & Body Co., South Main Street, *Concord, N. H.*

KENDALL, ALLEN, district manager, Hercules Motor Mfg. Co., *Canton, Ohio.*

KLEI, HERBERT E., student, Tri-State College, *Angola, Ind.*

KOLETSCHKE, HERBERT, chemist, Curtiss Aeroplane & Motor Corporation, *Buffalo, N. Y.*

LASLEY, DANA M., sales manager, motor division, Wellman-Seaver-Morgan Co., *Akron, Ohio.*

LEROY, OWEN, designer, tractor works, International Harvester Co., 2600 West Thirty-first Boulevard, *Chicago, Ill.*

LEVENE, BERNARD NORMAN, secretary and service manager, Levene Motor Co., 2200 Diamond Street, *Philadelphia, Pa.*

LUMET, GEORGES, director, Labatoire de l'Automobile Club de France, Boulevard Bourdon 80, *Neuilly, France.*

MCFARLAN, A. H., president, McFarlan Motor Co., *Connersville, Ind.*

MCLERIE, A. G., general manager and pilot, McCarthy Aero Service, Ltd., *New Liskeard, Ont., Canada.*

MCLERIE, ROY, assistant director of mechanical transport, Soldiers Civil Re-establishment Department, *Ottawa, Ont., Canada.*

MAAS, FRANK W., advertising manager, *Implement & Tractor Trade Journal, Kansas City, Mo.*

MAGILL, HERBERT L., designing engineer, truck department, tractor works, International Harvester Co., *Chicago, Ill.*

MARTIN, FRANK H., manager, Stewart-Warner Speedometer Corporation, *Chicago, Ill.*

MARTIN, J. R., JR., tool designer, Lincoln Motor Co., *Detroit, Mich.*

MAYNARD, H. E., production engineer, Oakland Avenue Plant, Maxwell Motor Co., Inc., *Detroit, Mich.*

MERRITT, RALPH V., general foreman of machine shop, Brewster & Co., Inc., *Long Island City, N. Y.*

MEYER, J. F., chief engineer, McFarlan Motor Co., *Connersville, Ind.*

MORGAN, A. L., engineer, Mapleleaf Mfg. Co., Ltd., 999 Christopher Columbus Street, *Montreal, Que., Canada.*

MOSES, ADOLPH, mechanical engineer, L-W-F Engineering Co., *College Point, N. Y.*

MOUGLY, HARRY C., chemist, Dayton-Wright Division of General Motors Corporation, 219 Indiana Avenue, *Dayton, Ohio.*

NEUMANN, ALAN B., assistant chief tester, tractor works, International Harvester Co., *Chicago, Ill.*

NOAKES, WILLIAM E., automotive engineer, engineering section, service division, Motor Transport Corps, *Washington.*

NOE, SHEROD S., chief engineer, Tool & Auto Products Co., East Seventy-third Street and St. Clair Avenue, *Detroit, Mich.*

NOWAK, W., designing engineer, Available Truck Co., 1539 North Kilpatrick Avenue, *Chicago, Ill.*

PARTRIDGE, A. G., vice-president, Firestone Tire & Rubber Co., *Akron, Ohio.*

PERRY, EDWARD D., chief draftsman, Watson Products Corporation, *Canastota, N. Y.*

PERSSON, O. F., designing engineer, General Electric Co., *Erie, Pa.*

PFEIFFER, RUDOLPH S., chief experimental tester, Holt Mfg. Co., *Peoria, Ill.*

PIERCE, GEORGE W., vice-president, Pierce Governor Co., *Anderson, Ind.*

REESE, WILLIAM D., research department, Fifth Avenue Coach Co., 10 East 102nd Street, *New York City.*

ROBERTS, EUGENE D., vice-president, Puget Sound Iron & Steel Works, *Tacoma, Wash.*

RUSSELL, CHARLES E., chassis designer, Dayton-Wright Division of General Motors Corporation, *Dayton, Ohio.*

RUTLEDGE, LESLIE CHARLES, motor engineer, Post Office Department, *Wanganin, N. Z.*

RYTTENBERG, H. R., branch manager, U. S. Tractor & Machinery Co., *Menasha, Wis.*

SCOTT, RALPH B., draftsman, Winton Co., *Cleveland, Ohio.*

SHINN, LORABEE C., mechanical draftsman, Indiana Truck Corporation, *Marion, Ind.*

SMITH, J. P., assistant manager of development department, Firestone Tire & Rubber Co., *Akron, Ohio.*

SORRICK, C. H., manufacturers' sales manager, Firestone Tire & Rubber Co., *Akron, Ohio.*

SOUTHARD, GEORGE H., JR., treasurer and general manager, J. & B. Mfg. Co., 65 Eagle Street, *Pittsfield, Mass.*

STANLEY, F. C., chief chemist, Raybestos Co., *Bridgeport, Conn.*

STOUGH, CHARLES R., chief draftsman, Hayes Wheel Co., *Jackson, Mich.*

STURM, H. P., general manager, Clarksburg Automobile Co., *Clarksburg, W. Va.*

SWIGERT, W. K., works manager, Oakes Co., *Indianapolis, Ind.*

TALLMAN, FRANK O., salesman, Standard Sales & Service Co., 1806 South Michigan Avenue, *Chicago, Ill.*

THOMAS, FIRST-LIEUT. ROBERT V., Motor Transport Corps General Depot, *Chicago, Ill.*

THORNBURG, H. A., student, 427 North Salisbury Street, *West Lafayette, Ind.*

VAN SCOTER, JOSEPH W., heat treater, Curtiss Aeroplane & Motor Corporation, *Buffalo, N. Y.*

WALKER, W. T., president and general manager, Powrlok Co., 1107 East 152nd Street, *Cleveland, Ohio.*

WARREN, R. M., Warren Brothers, *Sumter, S. C.*

WILLGOOS, ANDREW V. D., chief draftsman, Wright Aeronautical Corporation, *New Brunswick, N. J.*